HEAT TRANSFER TO WATER FLOWING ON AN INCLINED PLATE

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By

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ABSTRACT

Liquid side heat transfer coefficients were determined for an inclined falling-film heater, using a brass plate as the heating surface. The heating surface was 6 3/4 inches wide, 28 3/4 inches long and 3/4 inch thick. The heat-receiving liquid was water, introduced onto the plate by means of a spray distributor. The heating medium was steam.

The heat transfer coefficients were determined at mass flow rates per unit breadth ranging from 1850 to 10,120 lbs./hr.-ft. for angles of inclination with respect to the horizontal between nine and ninety degrees. The general trend showed an increase in the heat transfer coefficient as the angle of inclination was increased and as the liquid rate was increased. The results were correlated within a maximum deviation of $\pm$ 18%, and with an average deviation of 7%, by the dimensional equation:

$$h = 87\ (\sin \phi)^{0.2}\ (\Gamma)^{1/3}$$

where

h is the coefficient of heat transfer between liquid and surface, Btu./hr.-sq.ft.-°F.

ϕ is the angle of inclination with respect to the horizontal, degrees.

Γ is the mass flow rate per unit bredth, lbs./hr.-ft.

For a vertical fall of liquid the above equation reduces to

$$h = 87\ (\Gamma)^{1/3}$$

This checks very well with an equation derived from one by McAdams[5] for the estimation of the heat transfer coefficient for a vertical fall of liquid:

$$h = 86.5\ (\Gamma)^{1/3}$$

INTRODUCTION

Until recently, application of the falling-film heater had been limited to the ammonia refrigeration industry. Engineers engaged in this field have long recognized the value of the falling-film, as used on vertical and horizontal shell and tube condensers, with their: (1) high overall coefficients of heat transfer (2) more effective utilization of condensing surface (3) lower first cost per square foot of heating surface (4) small floor space requirements (5) lower maintenance cost in that the tubes can be cleaned without the removal of the condenser heads and (6) greater economic flexibility in that large overloads can be handled with no increase in pressure against which the cooling water must be pumped.

The high overall coefficients of heat transfer obtained in falling-film ammonia condensers can be attributed to high liquid-side coefficients occasioned by the extremely thin layers of liquid into which heat is conducted and in which the liquid travels at a relatively high velocity.

In 1935 investigations were started by Bays and co-workers[2,3,6] which confirmed the result that high liquid side coefficients of heat transfer could be obtained in falling-film vertical tubular heaters. The coefficient was found to increase with increased liquid rate. Where water was the liquid used, the following equation was found to hold for an average film temperature of 190°F:

$$h = 120\ (\Gamma)^{1/3} \qquad (1)$$

Cooper[4] et al have described the flow characteristics of liquid under isothermal conditions in liquid layer form.

STATEMENT OF THE PROBLEM

So far, all investigations[1,2,3,6,9,10] have been concerned with liquid side heat transfer coefficients with two kinds of film type heaters. The two types are: (1) a wetted-wall column in which the liquid flows in layer form down the inside wall of the vertical tube, and (2) the trombone cooler in which the liquid heat-receiving medium flows in film form over the outside of horizontal tubes.

Certain phases of falling-film characteristics have not been investigated. It may at times be advantageous to use a heater or cooler inclined to the horizontal at an angle other than ninety degrees. In this case, it is reasonable to believe that the liquid side heat transfer coefficient will be some function of the angle of inclination, as well as the other factors which are already known, and this particular aspect consequently deserves investigation. The purpose of this investigation is to determine the effect of the angle of inclination on the liquid side heat transfer coefficient, and to correlate the data so as to provide the necessary knowledge for the design of equipment which might utilize an inclined falling-film heater.

APPARATUS

To have cooling water flowing as a film on a surface inclined to the horizontal at angles between zero and ninety degrees it was necessary to use a flat plate, and to make provisions for rotating it through the entire range. The plate was eight inches wide, thirty inches long, and 3/4 inch thick. It was made of half-hard engravers brass (composition:[7] 61.5% copper, 37% zinc, and 1.5% lead), having a thermal conductivity[8] of 68 Btu./hr.-sq.ft.-° F. per ft. at 68° F. Beneath the plate was attached a steam chest. A double flanged connection was used. (For details see Figures 1 through 7).

Since the temperature drop across the liquid film had to be measured, four thermocouples were inserted at various points just below the top surface of the plate, the average depth below the surface being approximately 1/32 inch. Copper-constantan thermocouples, constructed of 24 gauge wire, were used. The wires were fused together with a mercury arc. Each thermocouple wire was enameled and wrapped with cotton thread. Each set of wires was covered with a cotton braid treated with flameproof paint. Holes approximately 0.1 inch in diameter were drilled in from the nearest side of the plate, the top of the hole being approximately 1/8 inch below the surface of the plate. The holes extended into the plate for a distance of approximately 1 3/4 inches. Another hole, inclined at an angle of approximately thirty degrees to the first hole was drilled from the top surface of the plate until a continuous hole existed from the side to the top of the plate. The holes were drilled in this manner so that as little distortion as possible of the normal flow of heat would result. The center of each thermocouple hole on the top surface was located approximately two inches from the nearest side. The couples were spaced uniformly over the length of the plate. Soft solder was used for making contact between the bead of the thermocouple and the plate (see Figure 8).

To assure even distribution of the liquid flow over the plate, two spray distributors were used (see Figures 9 and 10). Distributor number one was used for flow rates up to one pound per second, after which distributor number two was used. These directed the liquid onto a four inch calming section attached to the upper part of the plate by two flat-head screws. The calming section had holes drilled and countersunk so that the screw head would be even with the surface. The plate had holes drilled and tapped for 8-32 screws. Holes on the plate were aligned with the holes in the calming section and were located between the restraining sides. A rubber gasket was placed between the calming section and the plate. This made the calming section sit about 3/16 inch above the plate. Soft solder was used to fill any cracks around the screw heads. For other details see Figures 5, 6, and 11.

Similarly, at the bottom of the plate, a receiving and directing trough, constructed in the form of a triangular prism, was attached. This allowed the water to be collected more easily.

The restraining sides of the plate were sealed with "Permatex #1." This held well for a period of time, but soon allowed some leakage.

The steam chest was insulated with approximately a $\frac{1}{4}$ inch thickness of asbestos.

A photograph showing the equipment in operation, while inclined to the horizontal at an angle of 56°, is given in Figure 12.

FIGURE I
END CROSS-SECTION
OF
HEATER-PLATE ASSEMBLY

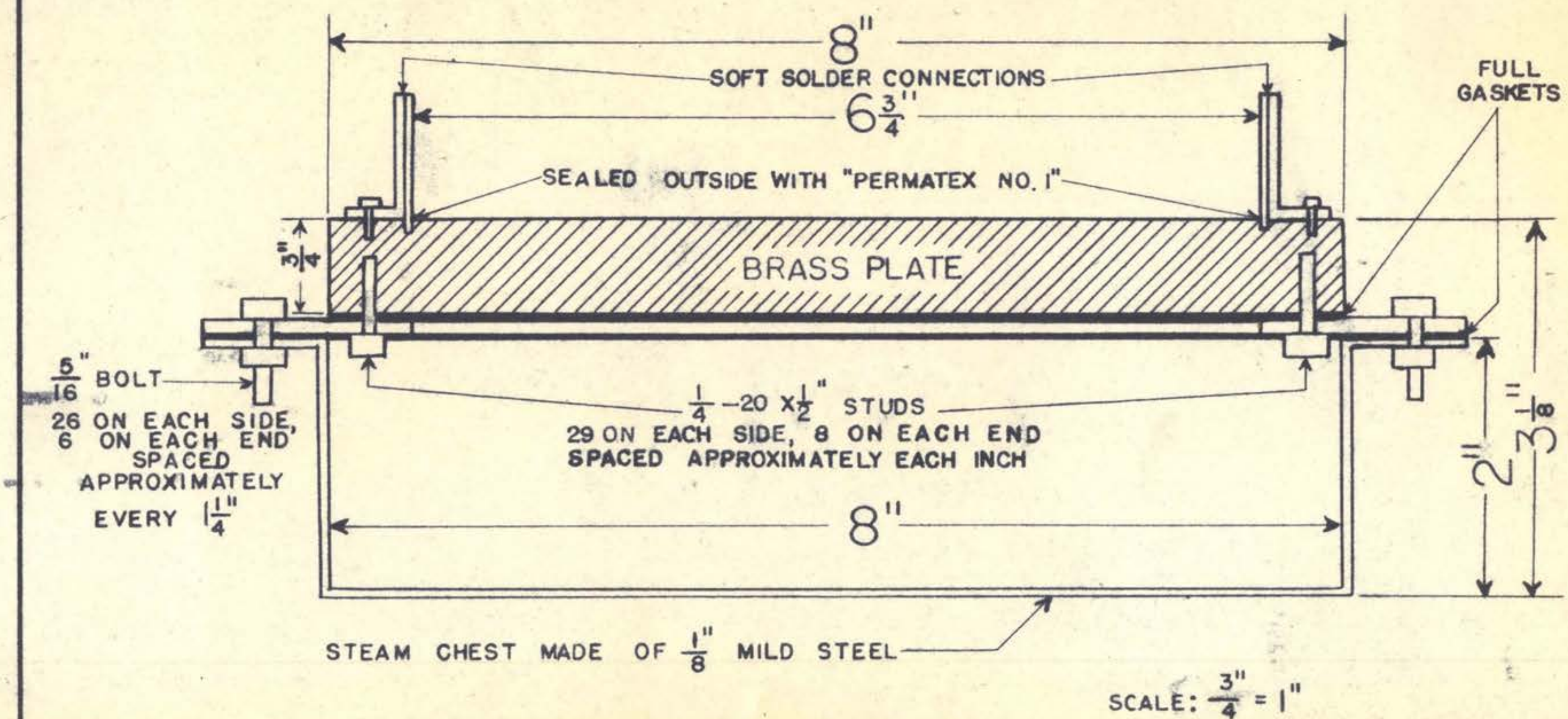
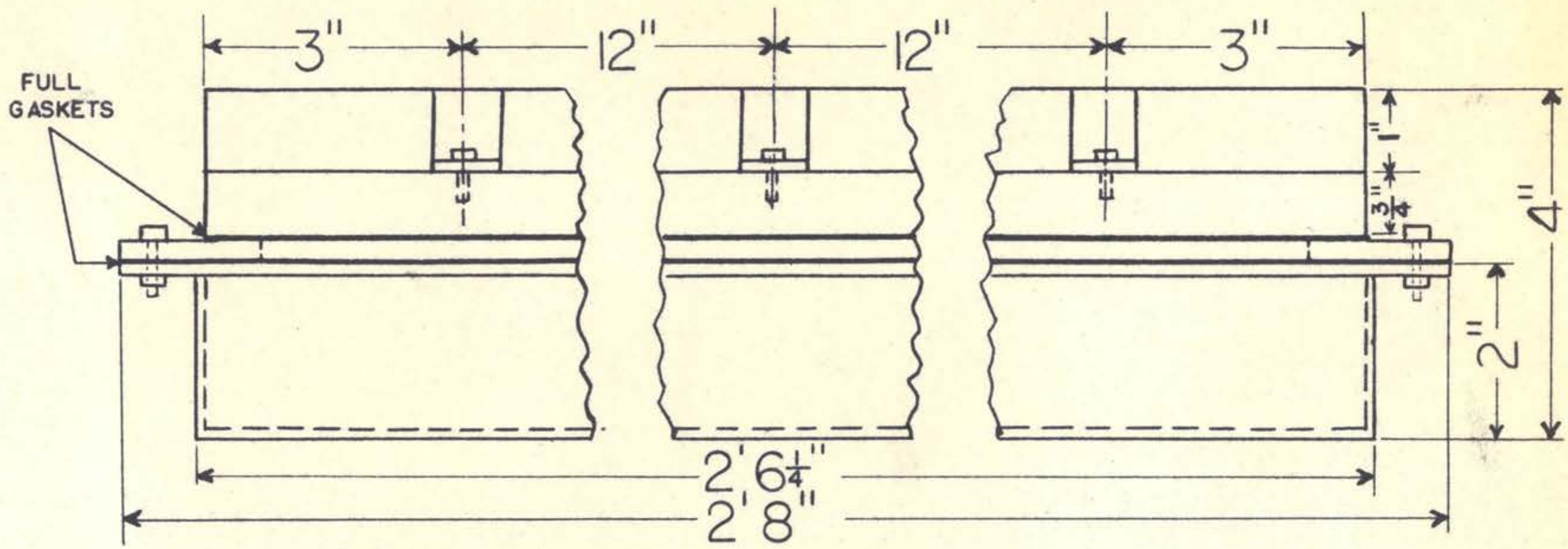


FIGURE 2
SIDE VIEW
HEATER-PLATE ASSEMBLY



SCALE: 1" = 2"

FIGURE 3
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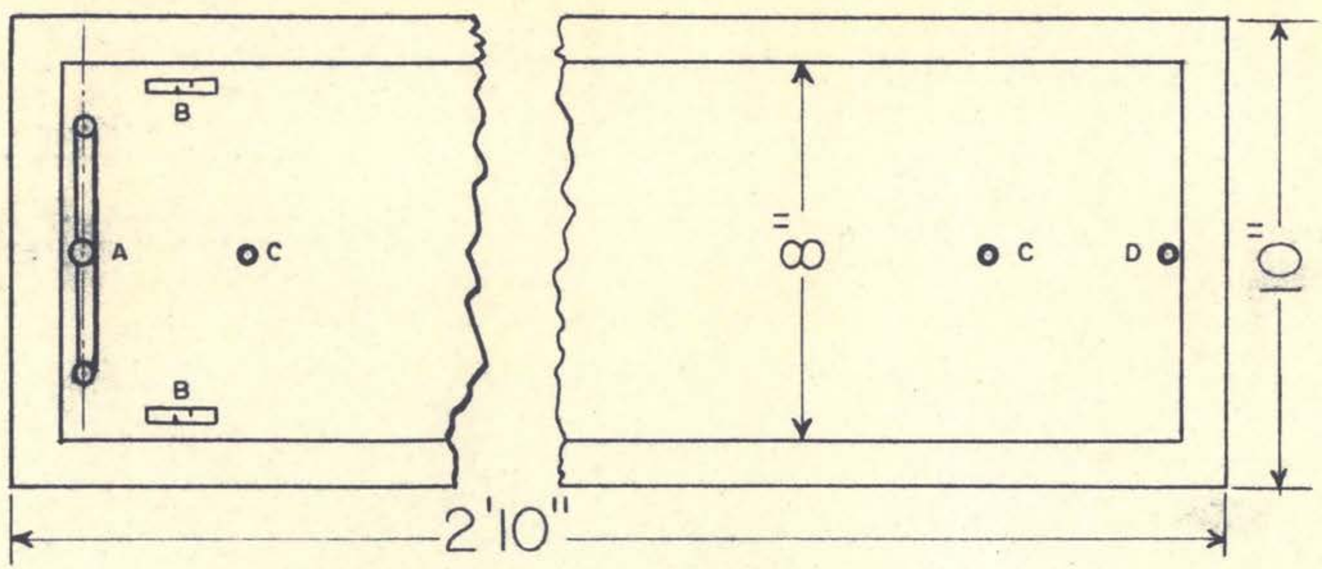
SHOWING
A. STEAM DISTRIBUTOR
B. HINGING MECHANISM
C. AUXILIARY INLETS TO STEAM CHEST
D. STEAM OUTLET



SCALE: 1" = 4"

Figure 4

Photograph of Bottom of Heater-Plate Assembly

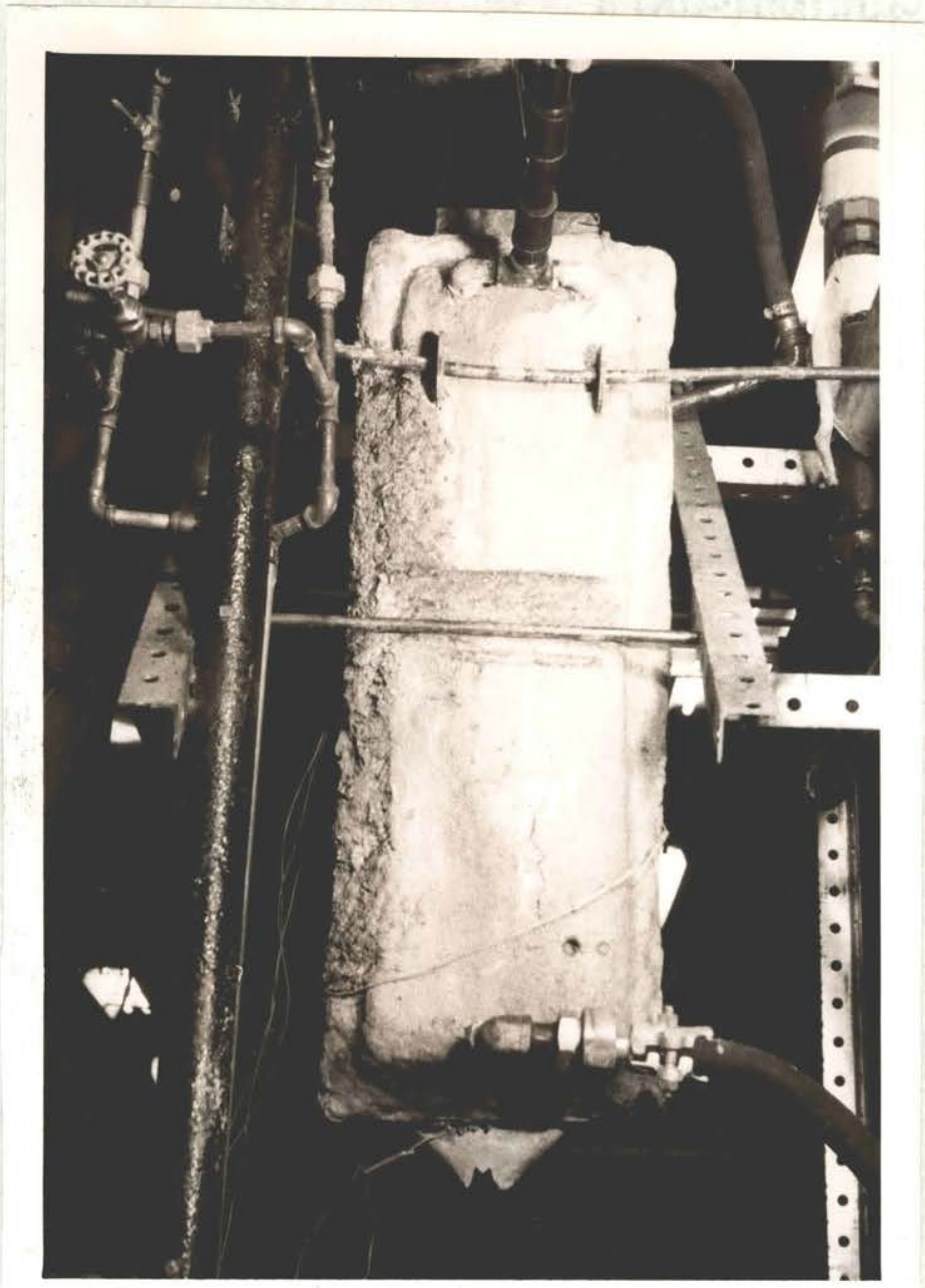


FIGURE 5
TOP VIEW
HEATER-PLATE ASSEMBLY

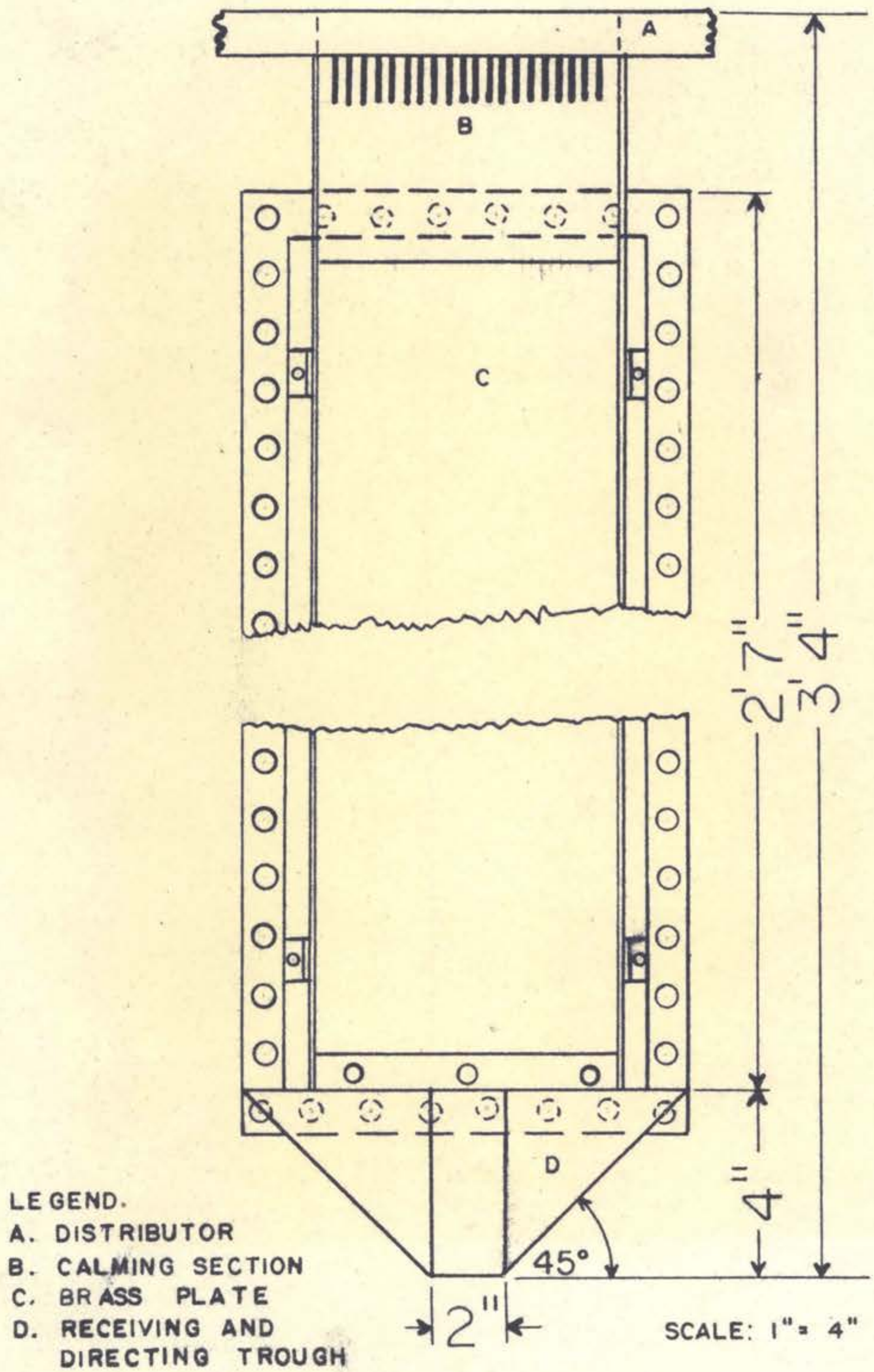


Figure 6

Photograph of Top of Heater-Plate Assembly

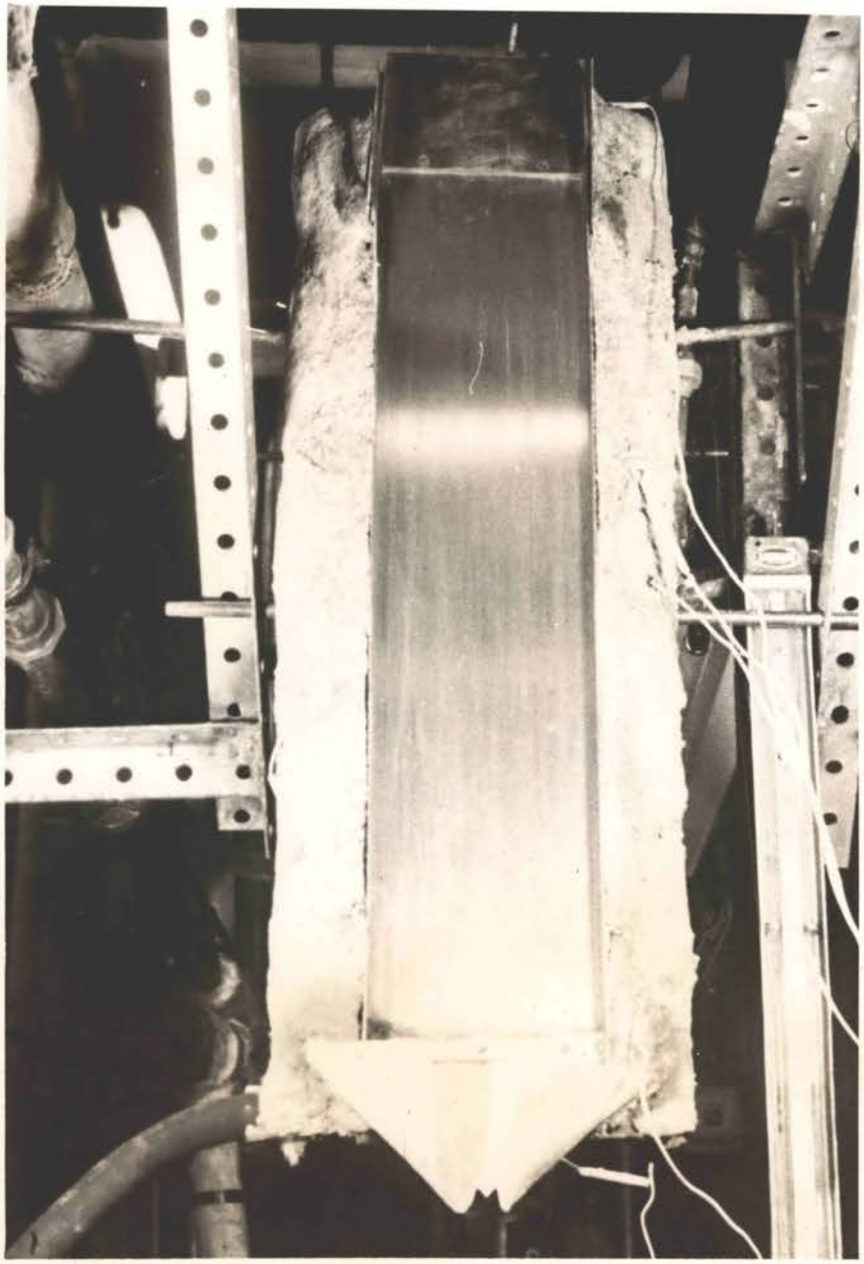


FIGURE 7
HINGING MECHANISM & SUPPORT
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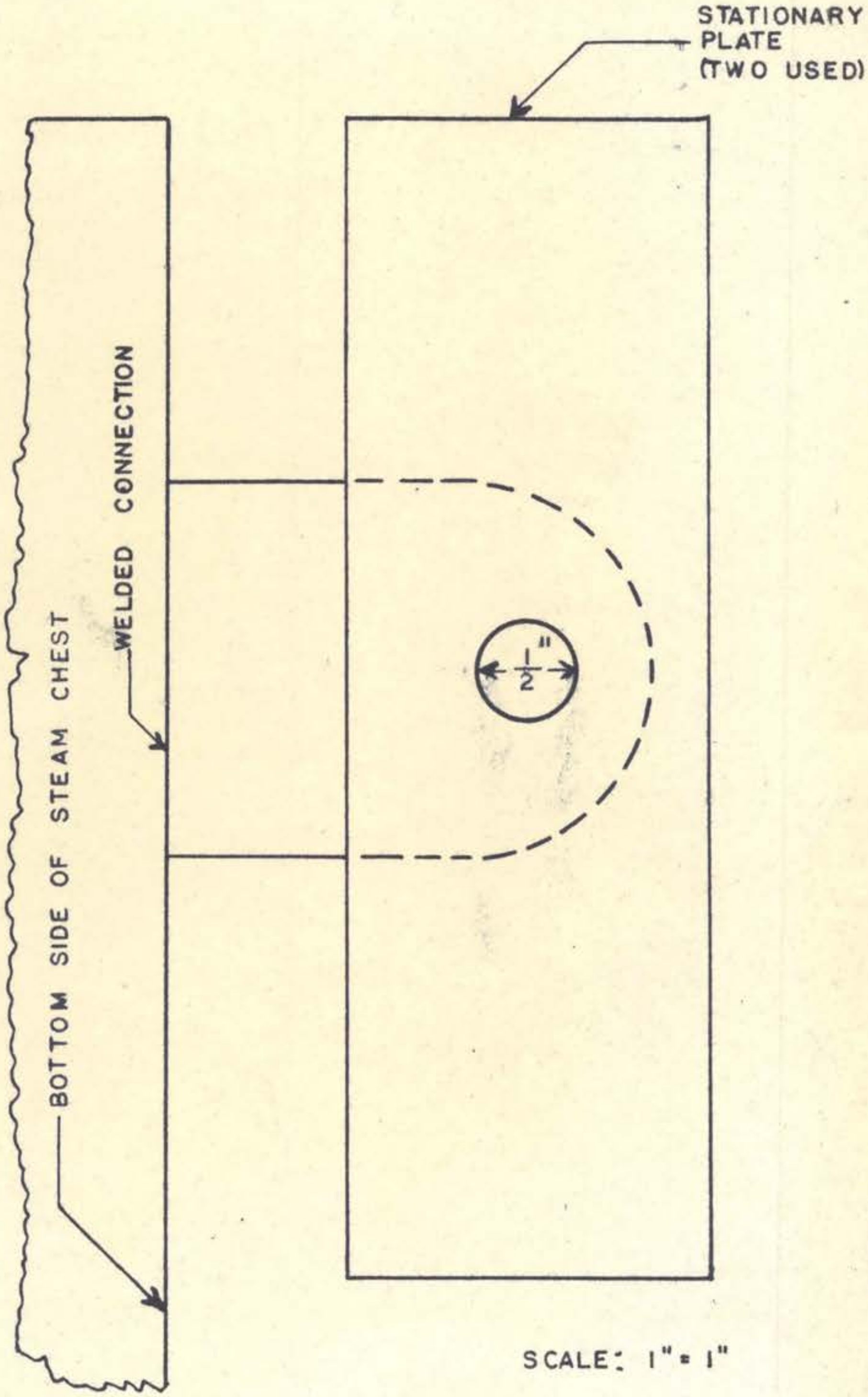

FIGURE 8
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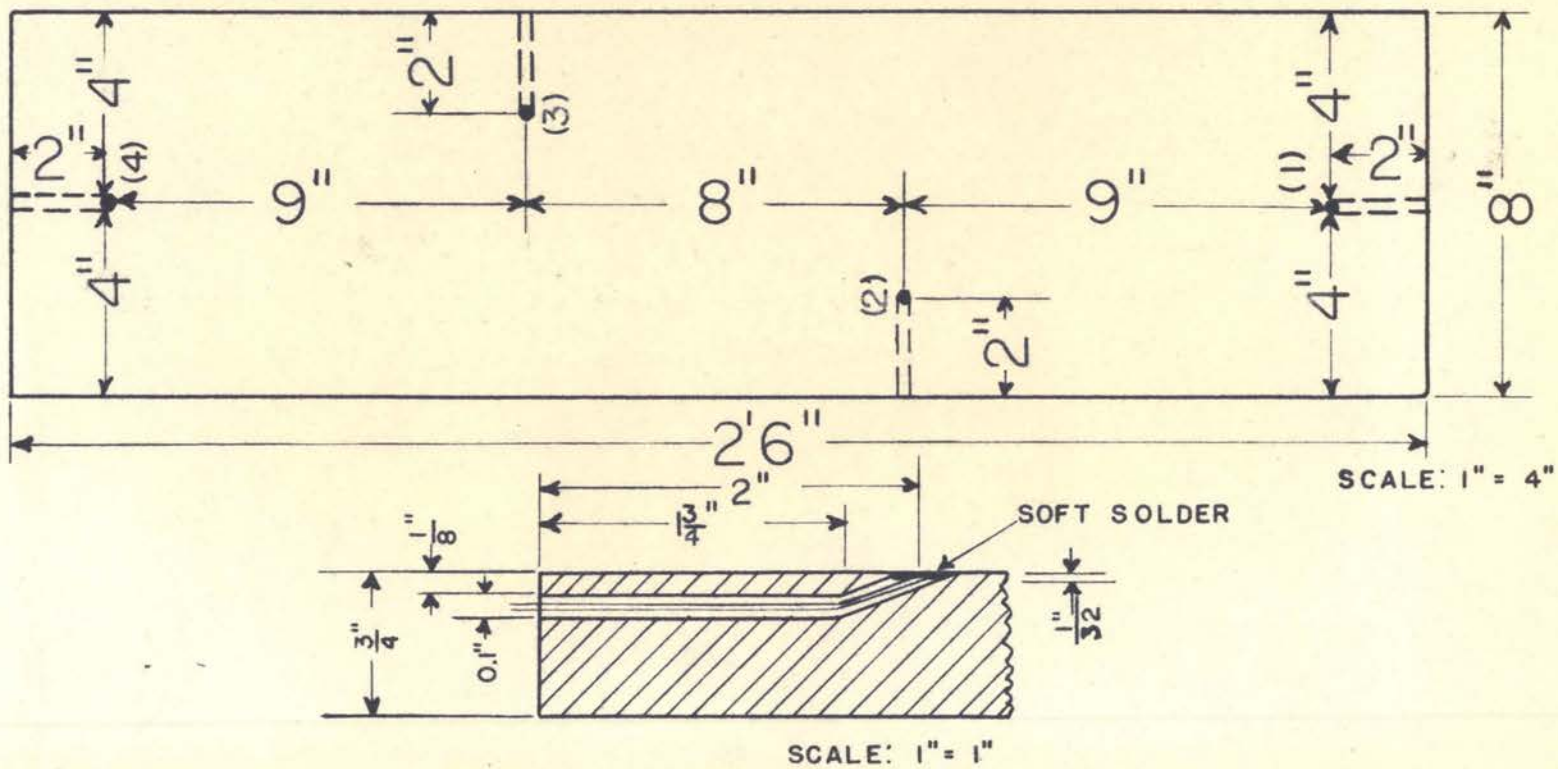
BRASS PLATE



NUMBER IN PARENTHESIS REFERS TO THE THERMOCOUPLE NUMBER

FIGURE 9
DISTRIBUTOR NO. 1
SECTIONAL VIEW

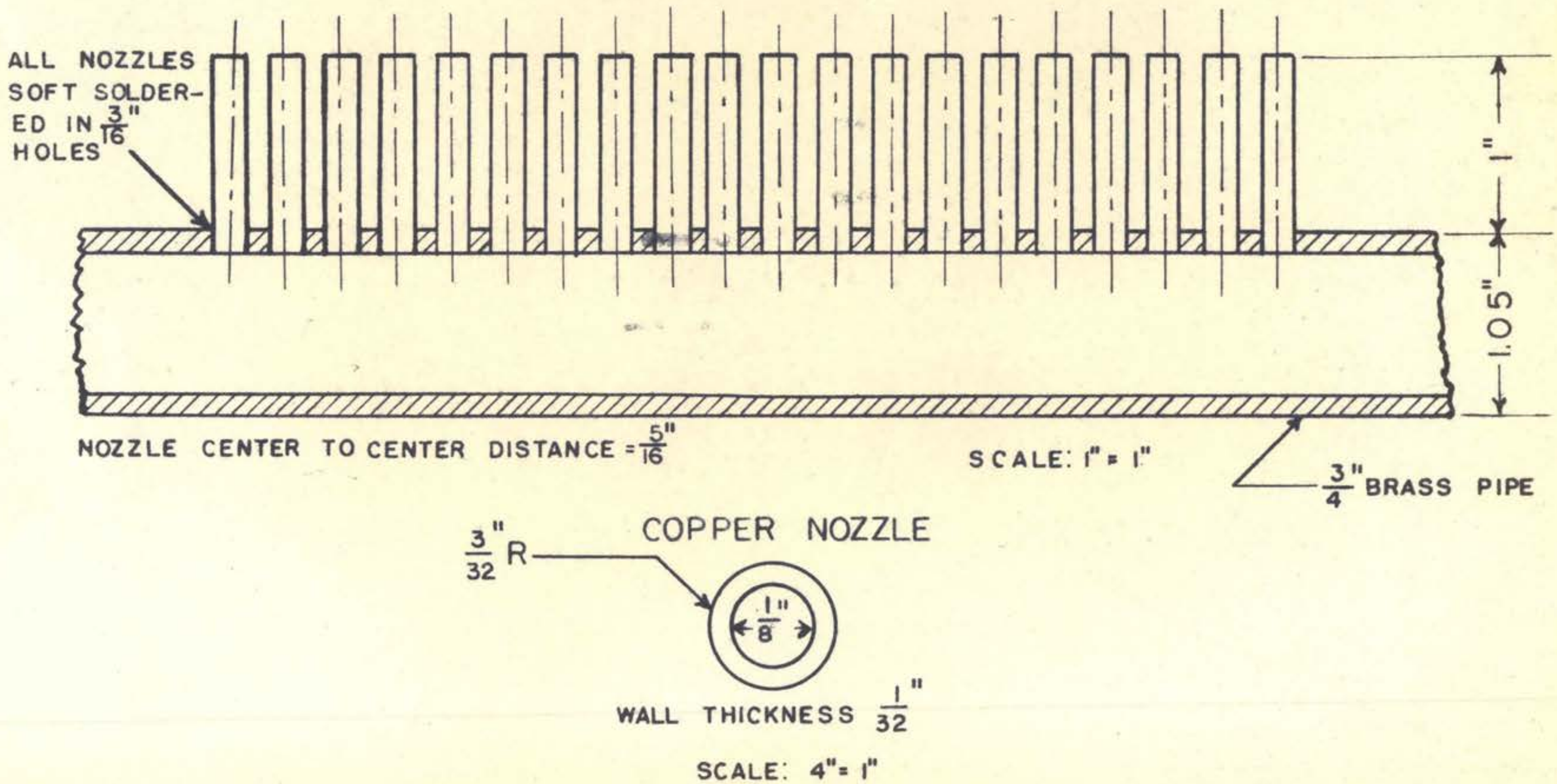


FIGURE 10
DISTRIBUTOR NO. 2
SECTIONAL VIEW

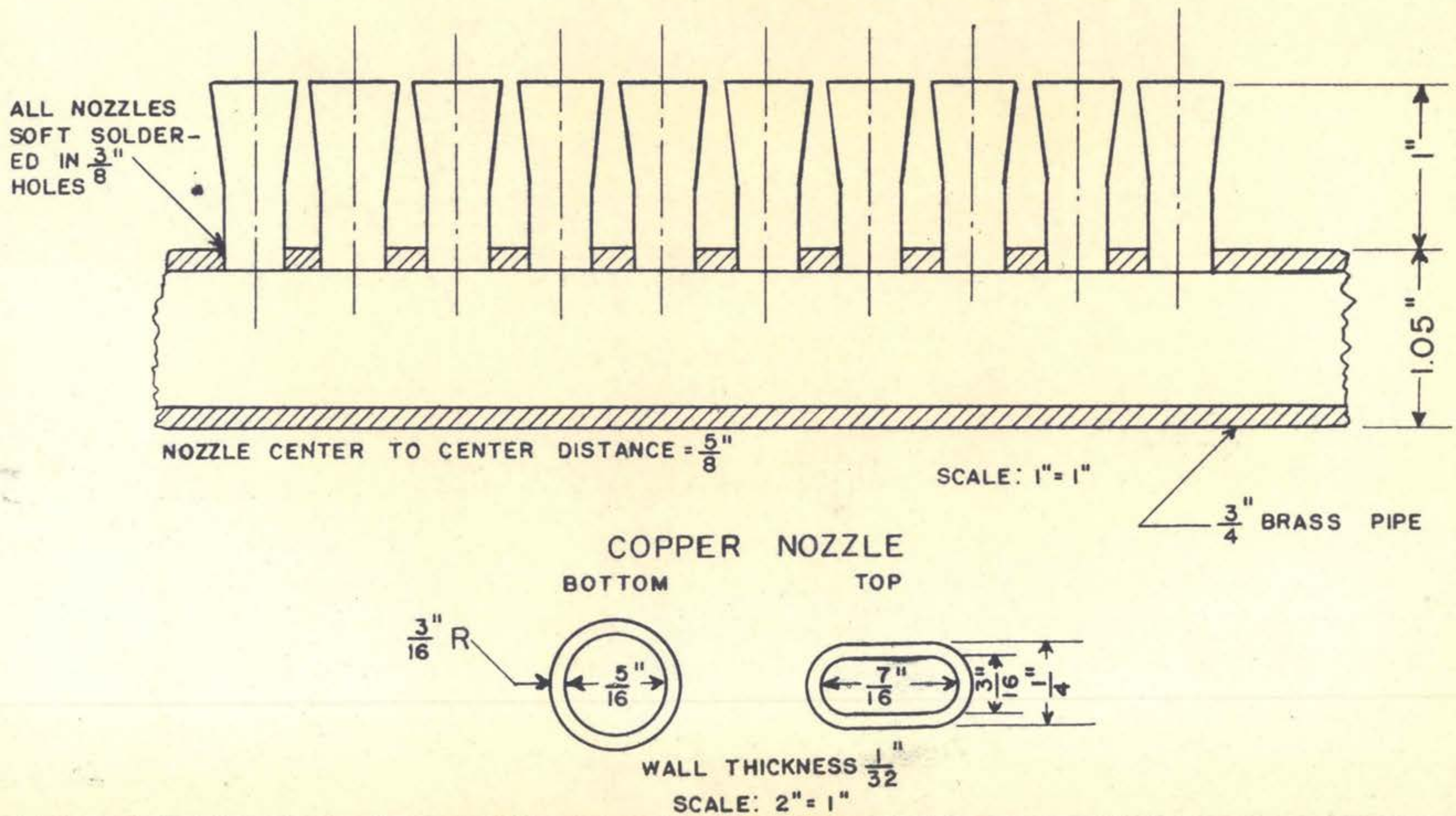


FIGURE 11
CALMING SECTION
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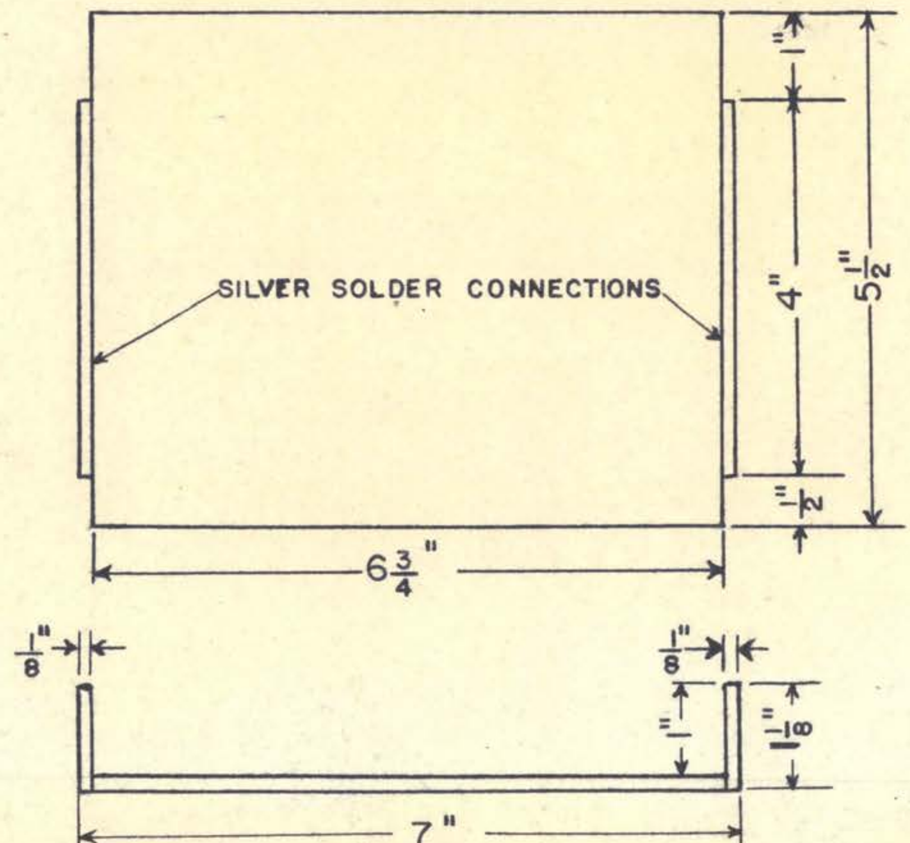
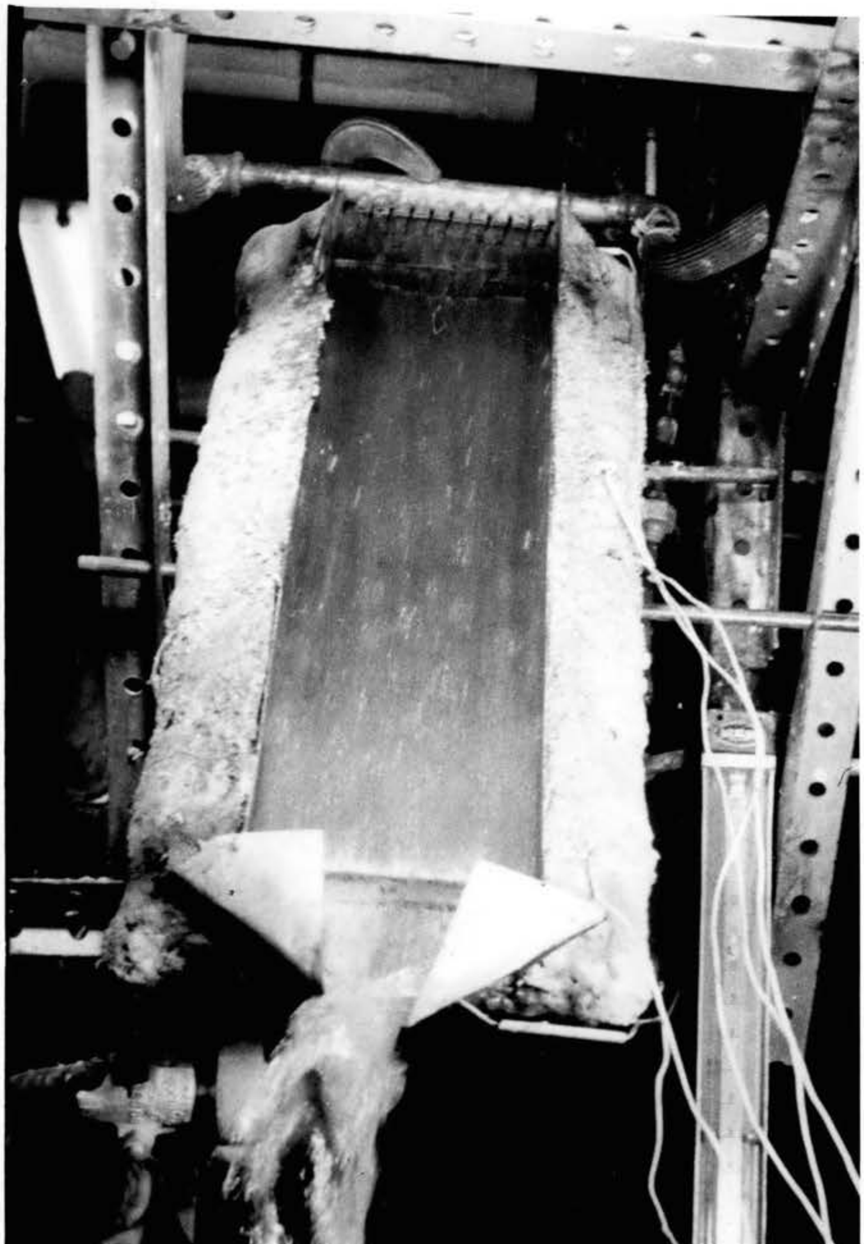


SCALE: 1"=2"

MATERIAL OF CONSTRUCTION—MILD STEEL

Figure 12

Photograph of Equipment in Operation



AUXILIARIES

The water supply was city tap water taken from a constant head overhead storage tank. A gear pump was installed in the supply line so that sufficient head could be obtained for high flow rates. The supply line was constructed so that the pump might be bypassed.

Two orifices were constructed for measuring the flow rate. The orifices were calibrated in place. One orifice was $\frac{1}{4}$ inch in diameter, giving flow rates ranging from zero to 0.5 pound per second. The other orifice was 0.9 inch in diameter, giving flow rates from 0.5 to 1.6 pounds per second. The pressure differential across the orifice was measured with a manometer, which gave differential readings of zero to thirty inches of carbon tetrachloride minus water. The orifices were installed in a ten foot section of standard two inch pipe. Distributor no. 1 was connected to the supply line by a two-foot length of 3/4 inch garden hose and the necessary pipe connections. The other end of the distributor was blocked off. Distributor no. 2 was connected to the supply line by two two-foot lengths of 3/4 inch garden hose and the necessary pipe connections. In this case water entered both ends of the distributor.

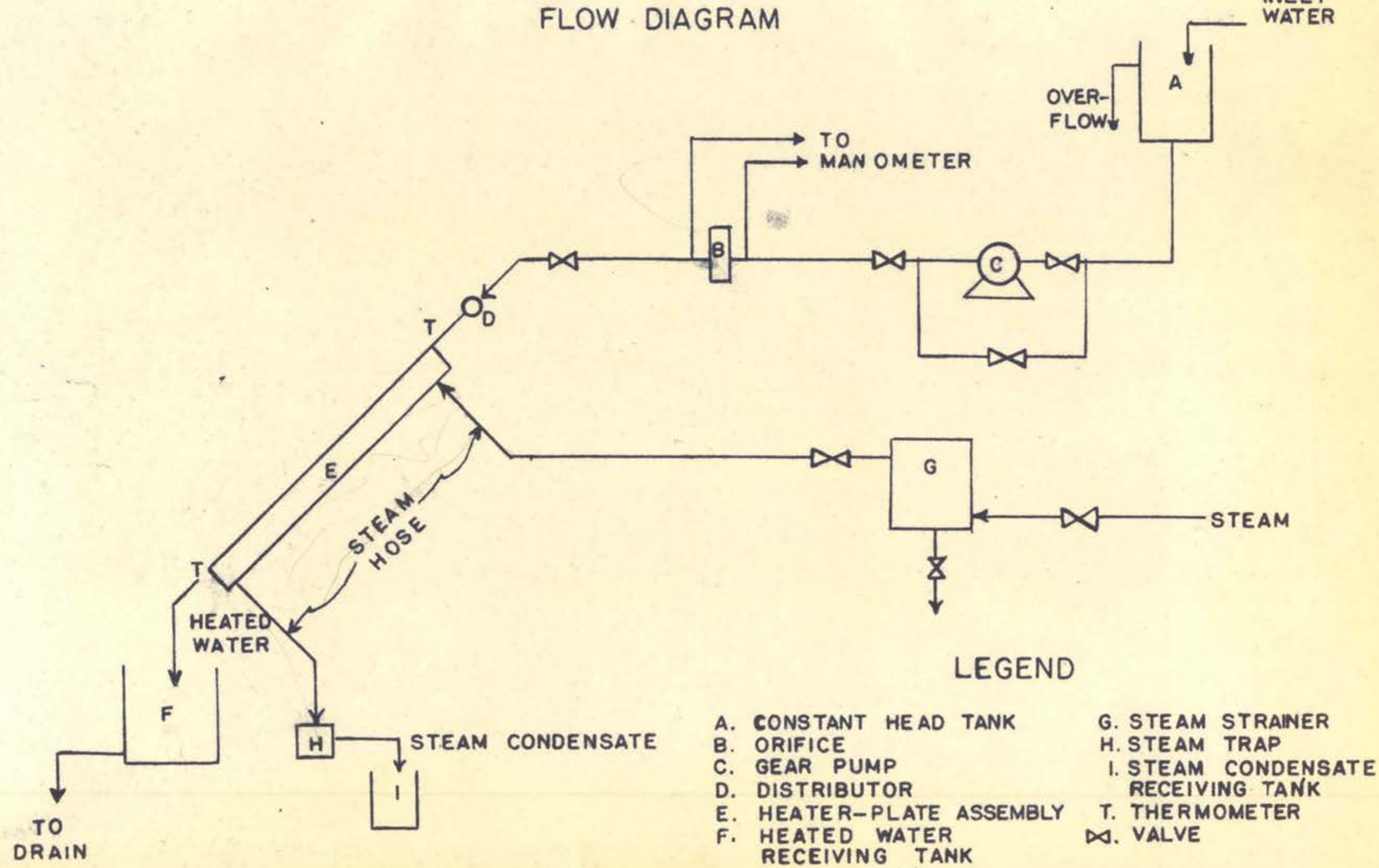
Saturated steam at five to ten pounds pressure was taken from the laboratory main. It passed through a steam filter filled with glass wool. While obtaining steady state conditions the steam pressure seldom varied more than 0.5 pound. The time required to reach steady state was approximately thirty minutes. Steam hose was used on both the inlet and outlet connections of the steam chest. A standard steam trap was installed in the outlet steam line.

Two 0 - 230° F. thermometers were used to measure the inlet and outlet temperatures of the water. These were inserted directly into the streams when it was necessary to obtain a reading.

The thermocouples were connected to a double pole switch, which allowed each thermocouple to be connected to a Leeds and Northrup portable potentiometer on a different circuit, i. e., there was no common lead. The potentiometer could be read accurately within ± 0.5 degree.

The outlet water discharged into a receiving tank which discharged to the drain. A flow diagram is given in Figure 13.

FIGURE 13
FLOW DIAGRAM



PROCEDURE

Calibration curves were made for the orifices, used for measuring the liquid flow rate, by observing the time required to collect 100 or 150 pounds (depending on the flow rate) of the liquid for a given differential. Flow rates, expressed as pounds of water per second, were calculated. The manometer reading is plotted versus the flow rate in Figures 24 and 25.

The two thermometers were calibrated at the freezing point and boiling point of water. They were read to 0.5 degree F. or less. No significant correction was required.

The thermocouples were calibrated in place, using the calibrated thermometers to determine the true temperature. Water at various temperatures flowed over the plate and through the steam chest simultaneously. Sufficient time was allowed for the plate to reach a uniform temperature. This was indicated by the thermometer inserted in the outlet stream reading the same as the one inserted in the inlet stream.

For temperatures near the boiling point of water and higher, the above calibration method is unsatisfactory. For such temperatures, steam at approximately 250 degrees F. was admitted to the steam chest and the plate allowed to reach equilibrium with the air. Using an estimated air film coefficient[5] of 1.0 - 1.5 Btu./hr.-sq.ft.-$^{\circ}$ F., the temperature drop across the plate to the bead of the thermocouple was calculated and found to be (within an estimated experimental error of 0.5 degree F.) with the observed temperatures.

In the making of a run the constant head tank was filled with water. The water rate was adjusted using the necessary valves and the pump, if required. The filter and steam lines were drained of any condensed steam, after which steam was admitted to the steam chest. The heater was inclined to the desired angle. The angle was measured with a protractor, using a plumb to obtain a vertical line.

After the readings indicated that steady state had been reached, the necessary data were taken. These included: flow rate as indicated by the monometer, inlet and outlet temperatures of the water, the angle of inclination with respect to the horizontal, and the temperature of the plate surface as indicated by the four thermocouples. The time required to reach steady state conditions was about thirty minutes.

In series XII to XVI, the water, as it left the calming section, sprayed onto the plate at a point sufficiently far down to necessitate making a correction for the heating area.

A set of runs was made at a particular flow rate, changing the angle of inclination after each run. Other sets of runs at different flow rates were made in a similar manner.

While operating, the nozzles of the distributor were clamped in place and inclined (approximately 10 to 15 degrees) downward with respect to the calming section. It was unnecessary to alter the position of the nozzles when the angle of inclination was changed.

In order to insure a clean heating surface throughout the investigation, the plate was lightly polished with number one emery paper before starting up after every shutdown.

CORRELATION OF RESULTS

The work of Bays indicated that liquid side heat transfer coefficients obtained in falling water films could be correlated by plotting on log-log paper the heat transfer coefficient, h, versus the water rate per unit breadth, Γ. This was done in this investigation for each of the angles studied. The best straight line was drawn through the points for each angle and the slope of each line determined. The slopes of the lines were as follows:

Angle of Inclination	Slope
18°	0.276
28°	0.296
42°	0.348
56°	0.401
67°	0.390
Average	0.342

The nine and ninety degree runs were not included for averaging because of the limited number of points on the curves. Since the average slope was very close to the value of 1/3 obtained by Bays, it was decided to take h as a function of $(\Gamma)^{1/3}$. The best lines of slope 1/3 were then drawn through the group of points for each angle. (See Figures 14 through 20). The coefficient, C, of the equation

$$h = C\,(\Gamma)^{1/3} \qquad (2)$$

was then determined for each angle.

The constants so obtained should be some function of the angle of inclination. They were plotted on log-log paper versus the sine of the angle of inclination. Quite a good straight line could be drawn through these points. The slope of the line was found to be 0.2 (see Figure 21). The equation for the line was found to be

$$C = 87\,(\sin \phi)^{0.2} \qquad (3)$$

giving the general equation for correlating all the results as:

$$h = .87\ (\sin \phi)^{0.2}(\Gamma)^{1/3} \tag{4}$$

All the points were then plotted on log-log paper in two ways:

1. h versus $(\sin \phi)^{0.2}(\Gamma)^{1/3}$, giving a straight line with a slope of unity (see Figure 22).
2. $h/(\sin\phi)^{0.2}$ versus Γ, giving a straight line with a slope of 1/3 (see Figure 23).

It can be seen that the derived equation lines fit the points quite well.

The value of h for each run was calculated from the correlating equation and compared with the experimental value. The data correlated within a maximum deviation of $\pm$ 18%, and with an average deviation of 7%.

The experimental data and calculated results are given in Tables I, II, and III in the Appendix.

DISCUSSION OF RESULTS

The work of Bays and coworkers[2,3,6] on vertical wetted-wall columns produced the following equation

$$h = 120\ (\Gamma)^{1/3} \tag{1}$$

for a mean film temperature of 190° F. When other conditions prevail, h may be estimated from an equation given by McAdams[5]

$$\frac{h}{(k^3p^2g/u_f^{\,2})^{1/3}} = 0.01(cu/k)^{1/3}(4\Gamma/u_f)^{1/3} \tag{5}$$

Equation (5) reduces to equation (1) at a mean film temperature of 190° F.

The mean film temperature of the present work was approximately 120° F. Evaluating equation (5) at this mean film temperature gives the equation

$$h = 86.5\ (\Gamma)^{1/3} \tag{6}$$

The results of this investigation were correlated by the equation

$$h = 87(\sin \phi)^{0.2}(\Gamma)^{1/3} \tag{4}$$

For a vertical fall of liquid equation (4) reduces to

$$h = 87\ (\Gamma)^{1/3} \tag{7}$$

Equations (6) and (7) check extremely well. Equation (1) correlated the data of Bays within $\pm$ 18%. Equation (4) correlated the data of the present work within a maximum of $\pm$ 18%, and with an average deviation of 7%, indicating that the same degree of accuracy prevailed in both investigations.

In a true falling-film the only energy the liquid possesses initially is its potential energy. The potential energy is equal to the product of the mass and the vertical distance available for its fall. In the case of an inclined plane the potential energy is equal to the product of the mass, the length of the inclined plane, and the sine of the angle of inclination with respect to the horizontal. For a given mass therefore the potential energy would be directly proportional to the sine of the angle of inclination, provided the length of the plane is constant. The length of the inclined plane remained essentially constant throughout this investigation. The velocity of flow of liquid down the plane is dependent upon the initial potential energy available. Since previous work has indicated that the heat transfer coefficient increases with increased flow, it was felt that the heat transfer coefficient would be a function of the sine of the angle of inclination. This proved to be the case.

At any given angle, the results indicated the heat transfer coefficient increases, as expected, with the flow rate (see Figures 11 through 17). However, in practically all cases, a temporary drop in heat transfer coefficient took place at a flow rate of 7100 lbs./hr./ft. The break was particularly pronounced at the lower angles of inclination. It was at this flow rate that distributor number one was replaced by distributor number two. It was necessary to change distributors since excessive spraying would have resulted at this flow rate had the use of distributor number one been continued. The heat transfer coefficient from this point on again increased regularly with increasing flow rate. The total area of the nozzles (20) of distributor number one was approximately 0.25 square inch. The total area of the nozzles (10) of distributor number two was approximately 0.75 square inch. Since the liquid velocity leaving the distributor is inversely proportional to the area, for the same mass flow rate, the liquid velocity leaving distributor number one would be three times the liquid velocity leaving distributor number two. For a flow rate of 7100 lbs/hr./ft. the liquid velocity leaving distributor number one would be 10.35 ft./sec, while the liquid velocity leaving distributor number two would be 3.45 ft./sec.

Higher distributor velocity would tend to produce a thinner film over the initial portion of the plate (that portion required for the liquid to attain its free flow velocity) and would therefore allow a greater amount of heat transfer. This effect was most noticeable at the lower angles of inclination. It is at these angles that the film thickness is relatively great at the upper end of the plate (hydraulic gradient effect) and the change in film thickness due to distributor velocity would be expected to be most noticeable.

This phenomenon may also be explained on a basis of the total energy the liquid possesses as it leaves the nozzles. The total energy would be equal to the potential energy plus the kinetic energy. For a given mass flow rate the initial kinetic energy remains constant regardless of the angle of inclination, while the potential energy is dependent on the angle of inclination. This clearly points out that for a given flow rate at lower angles of inclination, the kinetic energy obtained from the nozzle velocity becomes a greater portion of the total energy which the liquid possesses. Because of the increased energy due to the initial velocity the liquid would move at a higher velocity in a thinner film down the plate, and as pointed out previously, would allow a greater amount of heat transfer.

At some angles and high flow rates (greater than 7000 lbs./hr./ft.), the liquid, as it left the calming section, sprayed onto the plate at a point farther down the plate than the point where the number one thermocouple was located. Runs made under these conditions are specifically designated as such in Table III and the number one thermocouple readings so obtained were not used in calculating the average temperature of the plate. Also, it was necessary in these runs to make a correction for the actual heating area used in each case. This information is also given in Table III. For flow rates less than 7000 lbs./hr./ft. a constant area of 1.325 square feet was used. Any deviation from this area at the lower flow rates was less than 2% and no correction was made.

An inspection of the readings of thermocouple number four for runs XII B through G and XIV A through G showed them to be unusually high and out of line. This couple normally read the average of the other three. The cause of this behavior is not known, although thermocouple number four was located lower than the others and the leads were noticed to be wet with water on several occasions. These few questionable temperature readings were not used in obtaining the average temperature of the plate and are so marked in Table III.

The thermocouples, with the exception of number four, were located just below the surface. Thermocouple number four was right at the surface. A temperature correction was made for the estimated average distance the thermocouples were located beneath the surface. For details, see the sample calculation in the Appendix.

Relatively few runs were made at angles greater than 67 degrees, since at these angles water tended to spray excessively and the liquid film on the plate was less uniform.

If the plate were horizontal there would be some heat transfer. Equation (4) therefore does not hold down to zero degrees. It is likely that at some angle less than 18 degrees the heat transfer coefficient ceases to be a function of $(\sin \phi)^{0.2}$ and that natural convection phenomena become more important. This is indicated in Figure 21, as the value of C obtained at an angle of nine degrees lies somewhat below the line drawn.

The calming section was four inches long. While making runs it was noted that this was not sufficiently long to give absolute uniformity of flow over the complete length of the plate. More uniform flow would have resulted had an 18 to 24 inch calming section been used. Also, the calming section would have worked better had it been flush with the plate, instead of being 3/16 of an inch above the plate. Small pressure gradients in the 3/4 inch pipe feeding the distributor nozzles was another factor contributing to some initial non-uniformity of flow. In distributor number two water entered from both ends of the pipe and upon leaving the distributor the flow distribution seemed to be of a slight parabolic nature.

The amount of heat transferred was based on the amount of liquid and the temperature rise of the liquid flowing on the plate. Measurement of all temperatures were with $\pm$ 0.5 degree, with the exception of t_o which was measured to within $\pm$ 1.0 degree.

Γ is usually defined as the mass rate of flow per unit perimeter. As used in this work Γ was defined as the mass rate of flow per unit breadth. Had the first definition been used it would have necessitated measuring the film thickness and adding double that amount to the breadth of the plate. This error is negligible for a wide plate and moderate flow rates.

It was attempted to make blank runs to determine the amount of heat lost by radiation, so that steam consumption heat balance check might be made on each run. This attempt failed since there was some steam leakage.

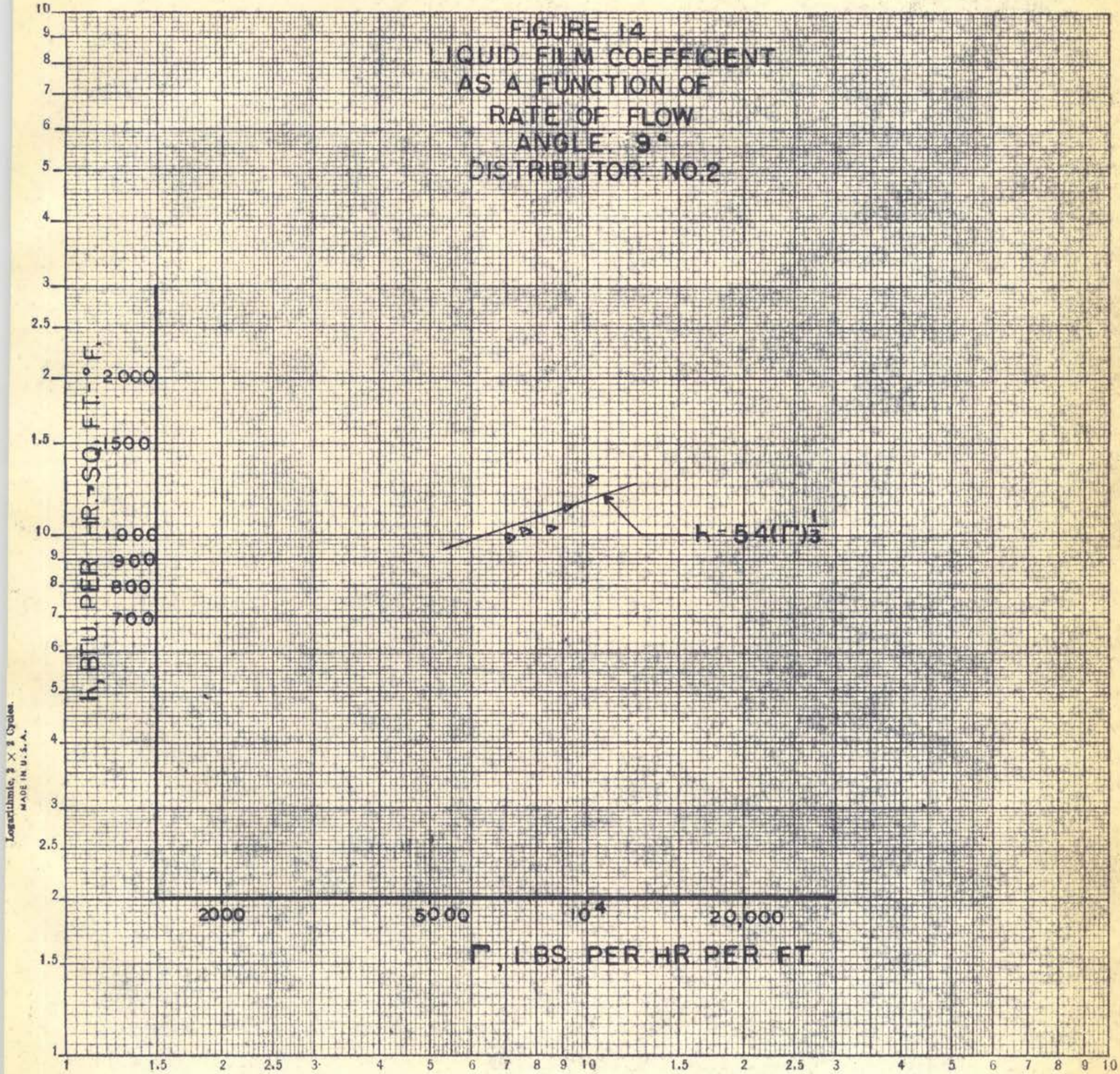


FIGURE 14
LIQUID FILM COEFFICIENT
AS A FUNCTION OF
RATE OF FLOW
ANGLE: 9°
DISTRIBUTOR: NO.2
h - 54(Γ)^1/3
h, BTU. PER HR.-SQ. FT.-°F.
2000
1500
1000
900
800
700
2000
5000
10^4
20,000
Γ, LBS. PER HR PER FT.
Logarithmic, 2 × 2 Cycles.
MADE IN U.S.A.

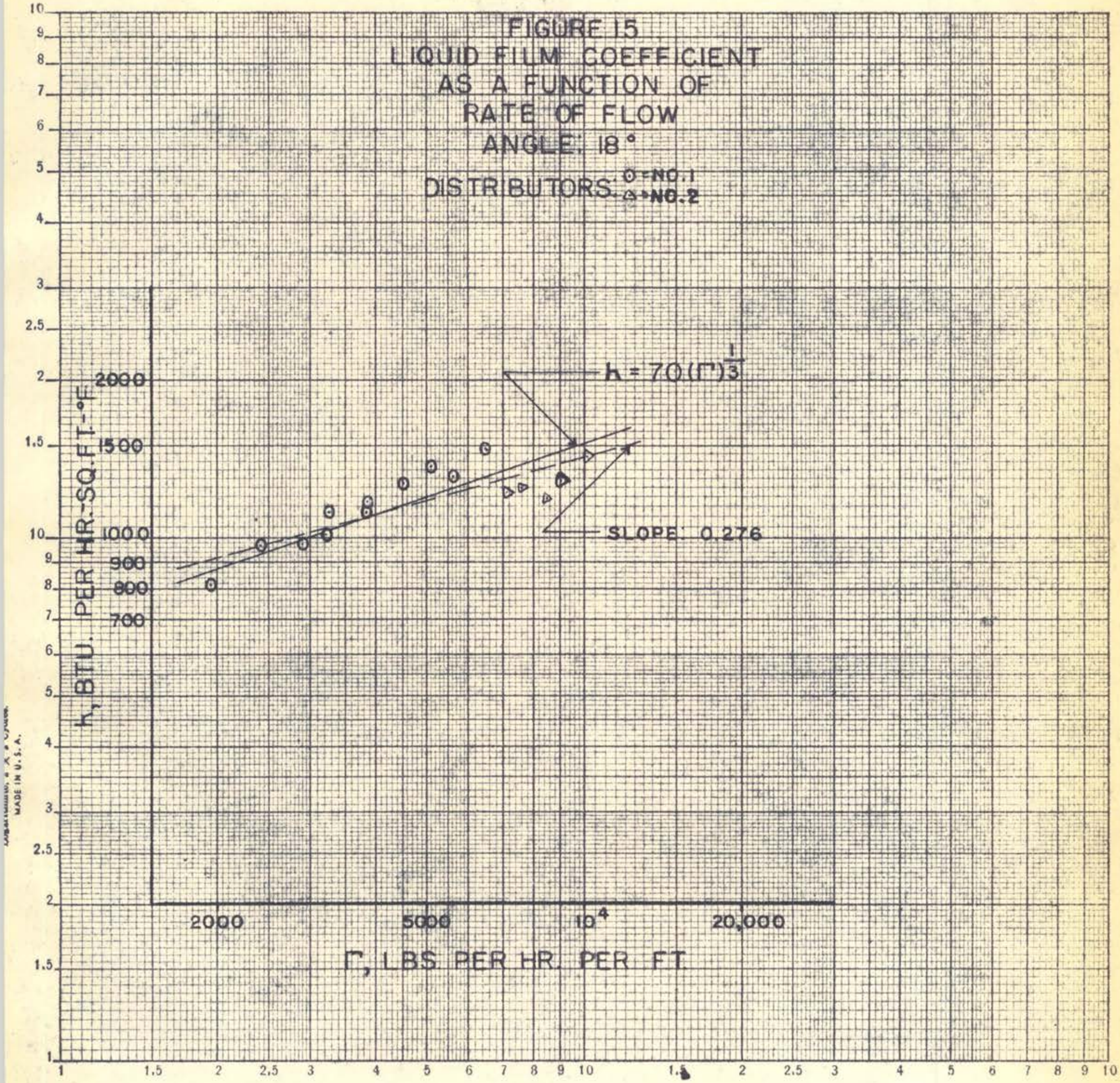


FIGURE 15
LIQUID FILM COEFFICIENT
AS A FUNCTION OF
RATE OF FLOW
ANGLE: 18°
DISTRIBUTORS: ⊙ = NO. 1
△ = NO. 2
h = 70(Γ)^(1/3)
SLOPE: 0.276
h, BTU. PER HR.-SQ.FT.-°F
2000
1500
1000
900
800
700
2000
5000
10^4
20,000
Γ, LBS. PER HR. PER FT.
MADE IN U.S.A.

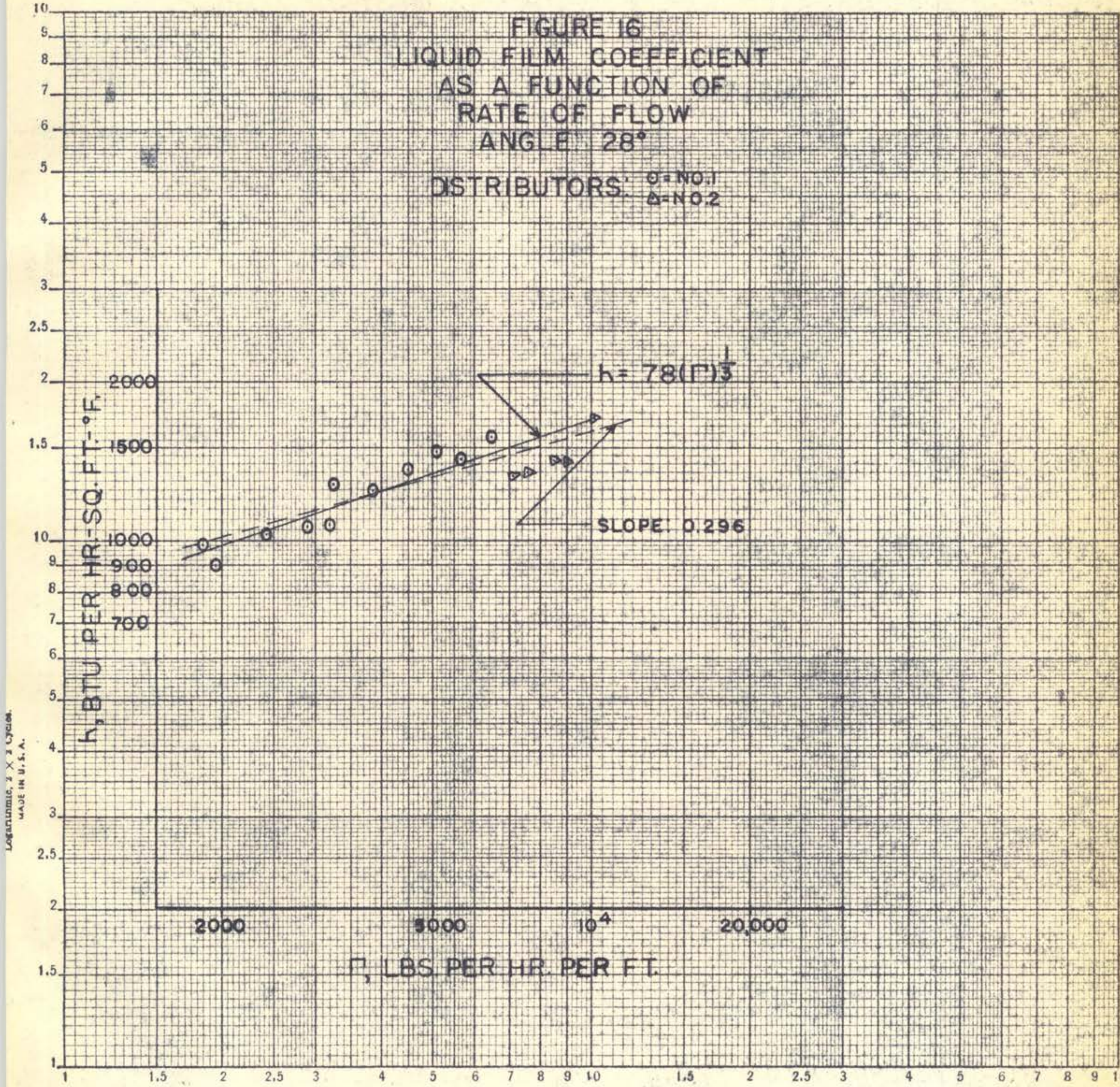


FIGURE 16
LIQUID FILM COEFFICIENT
AS A FUNCTION OF
RATE OF FLOW
ANGLE: 28°
DISTRIBUTORS: O=NO.1 Δ=NO.2
h= 78(Γ)^1/3
SLOPE: 0.296
h, BTU PER HR-SQ. FT.-°F.
2000
1500
1000
900
800
700
2000
5000
10^4
20,000
Γ, LBS. PER HR. PER FT.

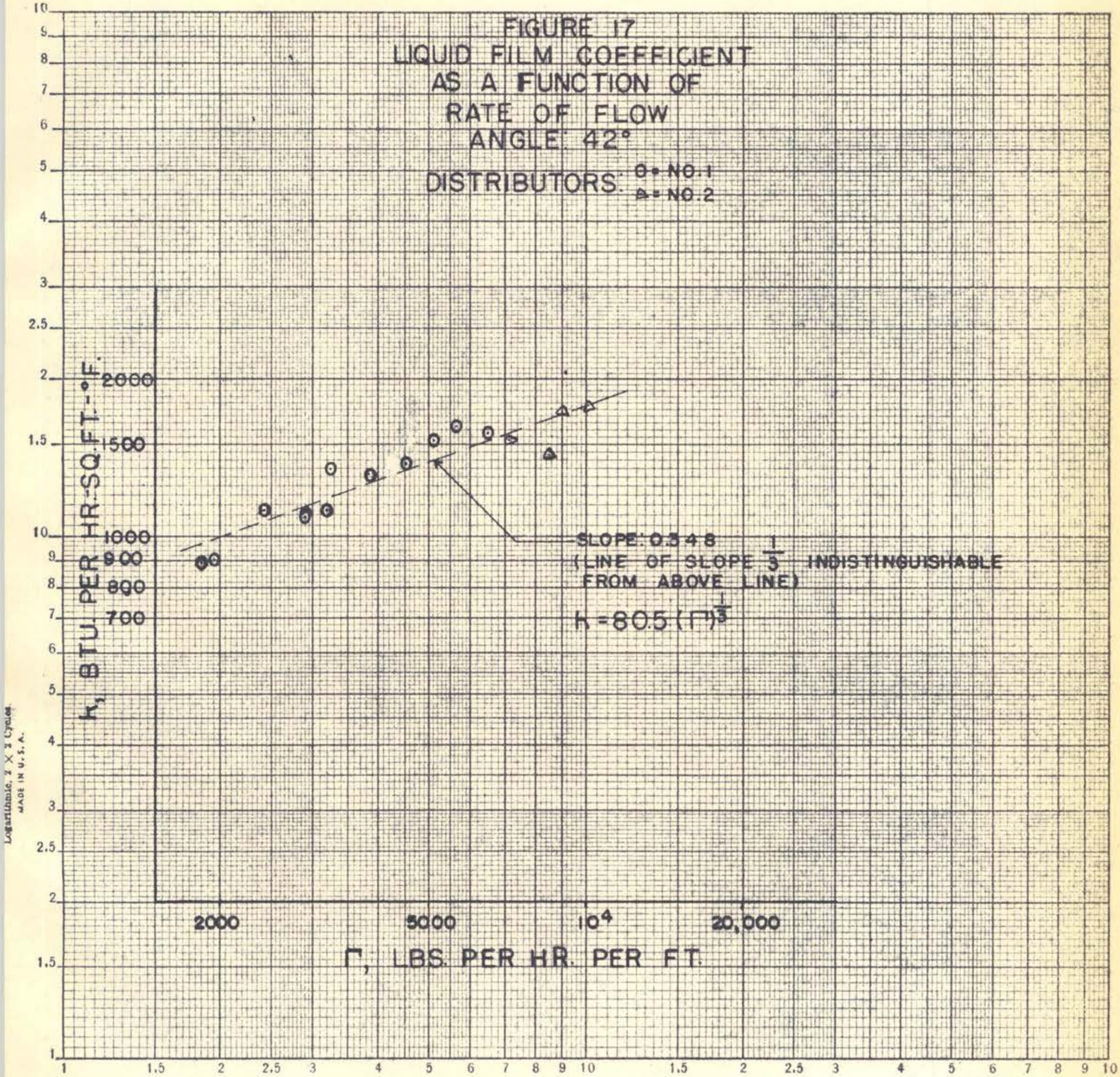


FIGURE 17
LIQUID FILM COEFFICIENT
AS A FUNCTION OF
RATE OF FLOW
ANGLE: 42°
DISTRIBUTORS: ⊙ = NO.1 △ = NO.2
SLOPE: 0.348
(LINE OF SLOPE 1/3 INDISTINGUISHABLE FROM ABOVE LINE)
h = 80.5 (Γ)^(1/3)
h, BTU. PER HR-SQ.FT.-°F.
2000
1500
1000
900
800
700
2000
5000
10^4
20,000
Γ, LBS. PER HR. PER FT.
Logarithmic, 2 X 2 Cycles
MADE IN U.S.A.

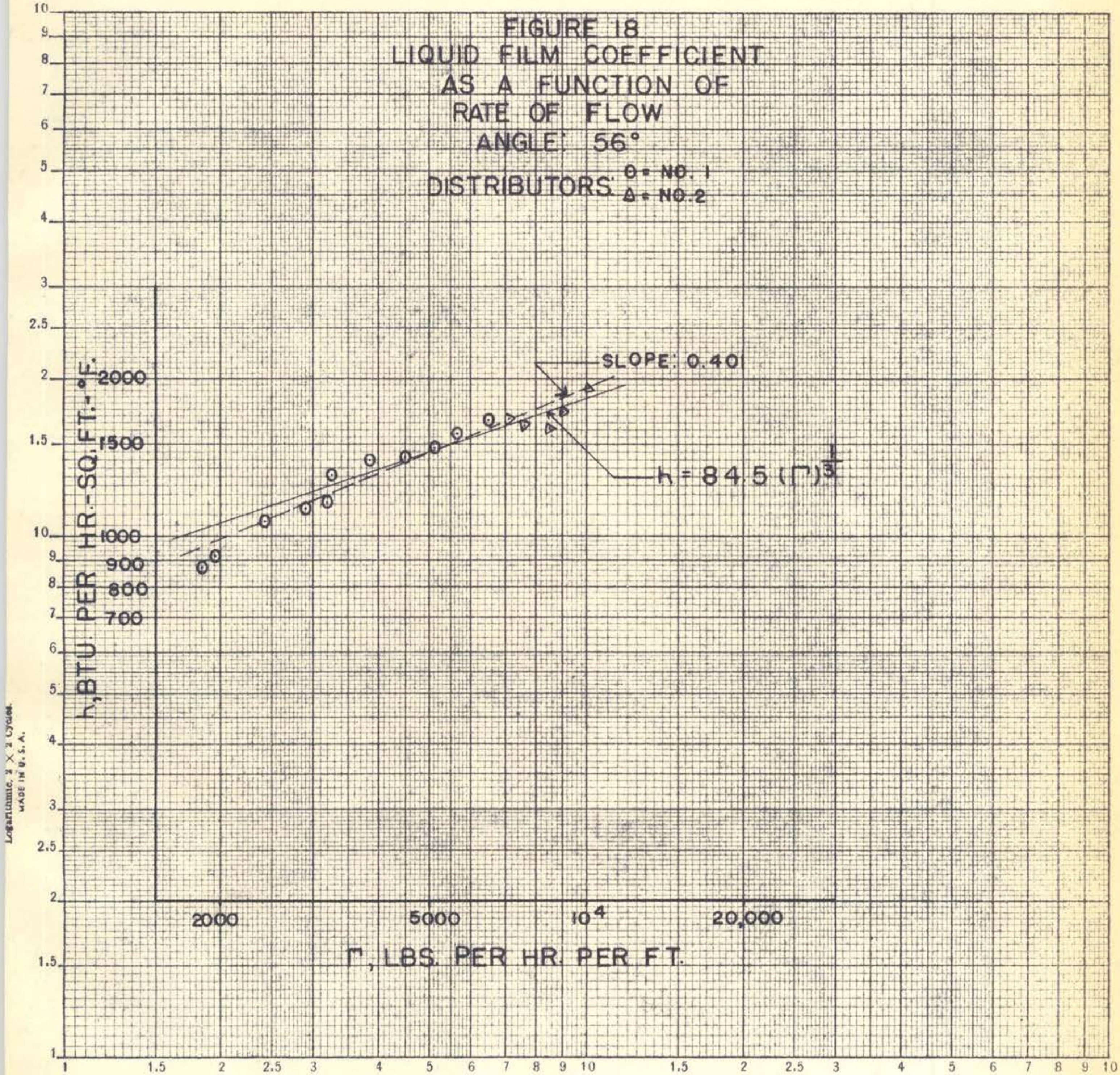


FIGURE 18
LIQUID FILM COEFFICIENT
AS A FUNCTION OF
RATE OF FLOW
ANGLE: 56°
DISTRIBUTORS: O = NO. 1
Δ = NO. 2
SLOPE: 0.401
h = 84.5 (Γ)^1/3
h, BTU PER HR.-SQ. FT.-°F.
2000
1500
1000
900
800
700
2000
5000
10^4
20,000
Γ, LBS. PER HR. PER FT.

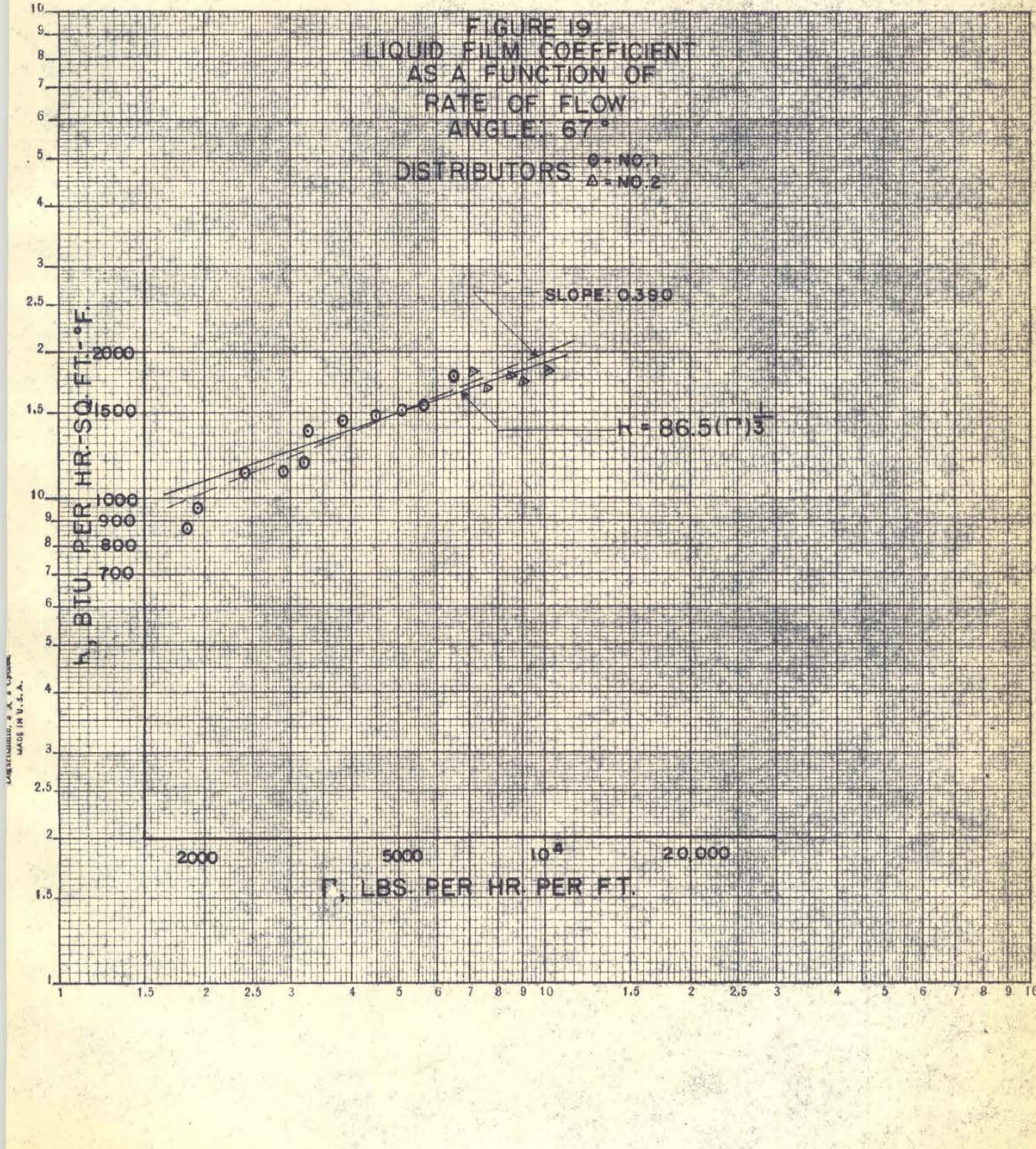


FIGURE 19
LIQUID FILM COEFFICIENT
AS A FUNCTION OF
RATE OF FLOW
ANGLE: 67°
DISTRIBUTORS ⊙ = NO. 1 Δ = NO. 2
SLOPE: 0.390
h = 86.5(Γ)^(1/3)
h, BTU. PER HR.-SQ. FT.-°F.
2000
1500
1000
900
800
700
2000
5000
10^4
20,000
Γ, LBS. PER HR. PER FT.

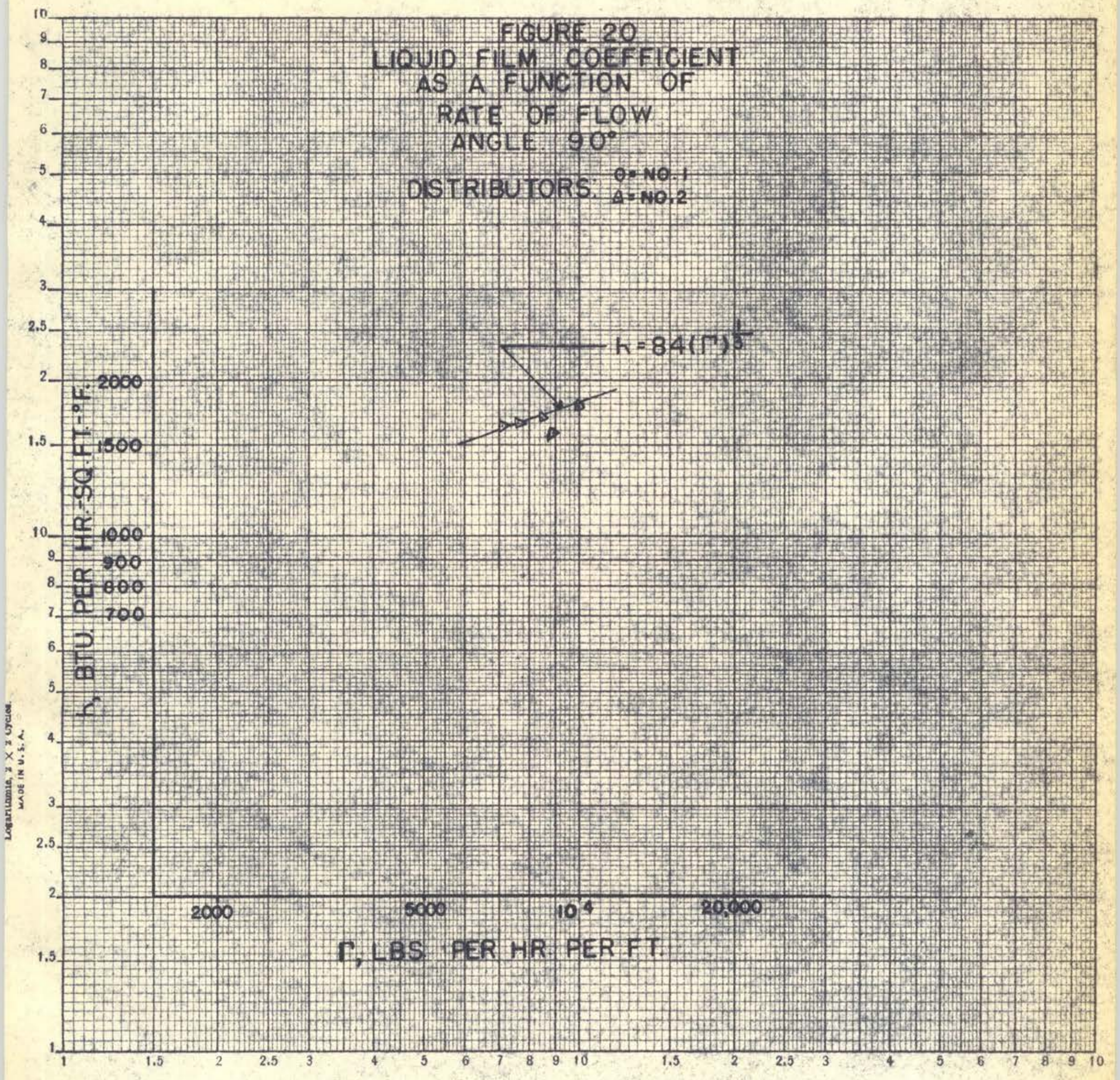


FIGURE 20
LIQUID FILM COEFFICIENT
AS A FUNCTION OF
RATE OF FLOW
ANGLE 90°
DISTRIBUTORS: O = NO. 1 Δ = NO. 2
h = 84(Γ)^(1/3)
h, BTU. PER HR.-SQ. FT.-°F.
2000
1500
1000
900
800
700
2000
5000
10^4
20,000
Γ, LBS. PER HR. PER FT.
Logarithmic, 2 X 2 Cycles.
MADE IN U. S. A.

FIGURE 21
$C = h/\Gamma^{1/3}$
AS A FUNCTION OF
SIN ϕ

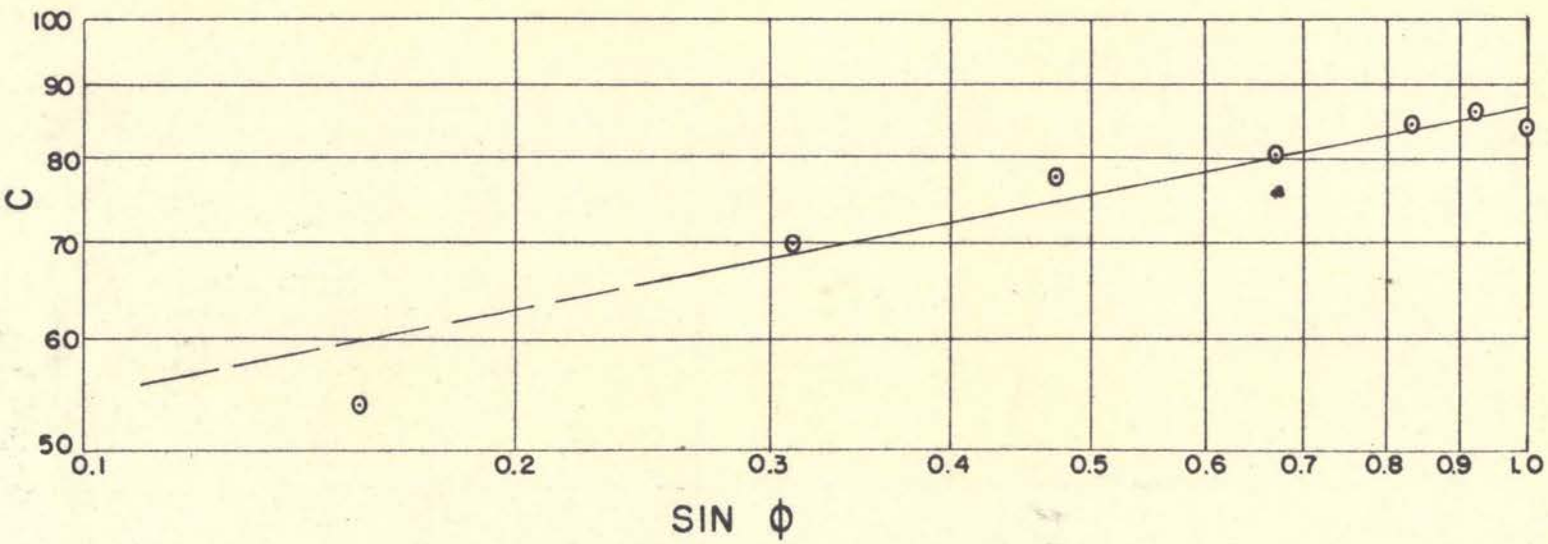
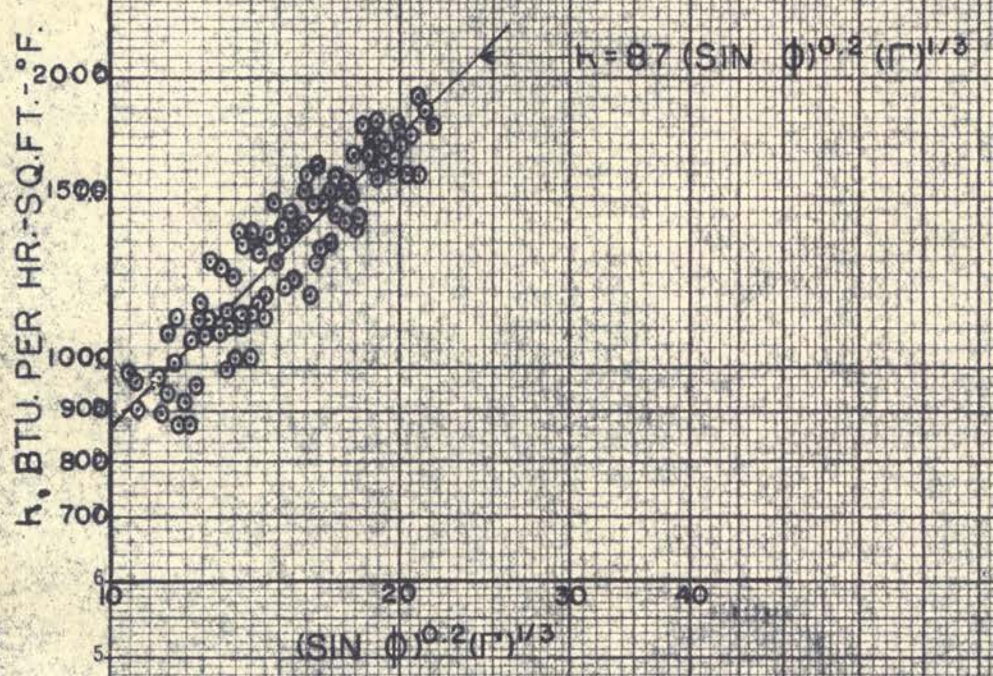


FIGURE 22
AGREEMENT OF DATA
WITH
GENERAL CORRELATING EQUATION
h VERSUS $(SIN\ \phi)^{0.2}(\Gamma)^{1/3}$



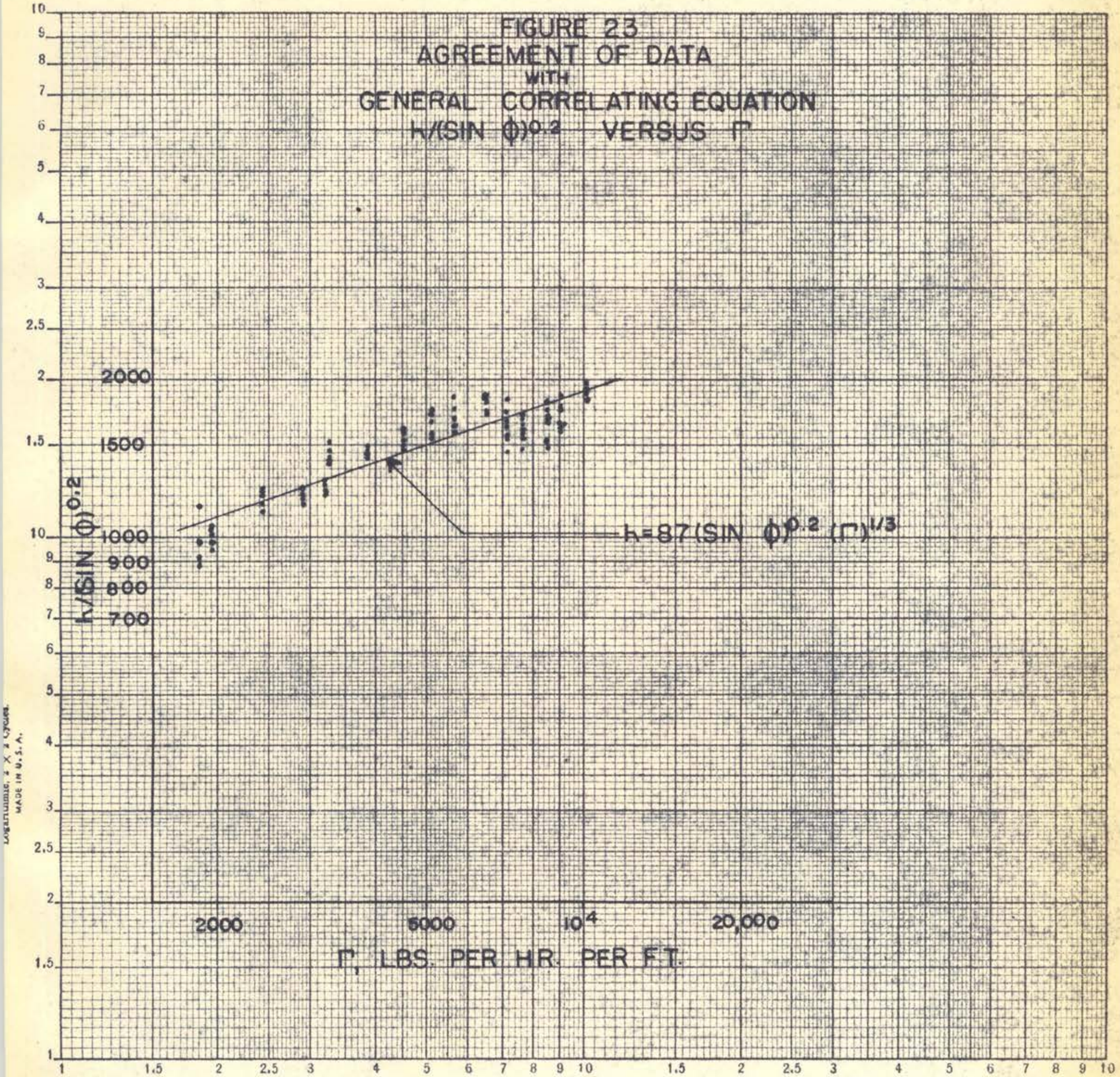


FIGURE 23
AGREEMENT OF DATA
WITH
GENERAL CORRELATING EQUATION
h/(SIN ϕ)^0.2 VERSUS Γ
h=87(SIN ϕ)^0.2 (Γ)^1/3
h/(SIN ϕ)^0.2
2000
1500
1000
900
800
700
2000
5000
10^4
20,000
Γ, LBS. PER HR. PER FT.
Logarithmic, 2 X 2 Cycles
MADE IN U.S.A.

SUMMARY

A study of water film heat transfer coefficients was made on an inclined plate heater, using steam as the heating medium.

A brass plate, capable of being inclined at various angles with respect to the horizontal, was used. The heating surface was 6 3/4 inches wide, 28 3/4 inches long and 3/4 inch thick. The water was introduced onto the plate by means of a spray distributor.

The heat transfer coefficients were determined at flow rates per unit breadth ranging from 1850 to 10,120 lbs./hr.-ft. for angles of inclination between nine and ninety degrees. All flow rates were in the turbulent range, with Reynolds numbers varying from 2900 to 12,800. The coefficients so obtained ranged from 817 to 1912 Btu./hr.-sq.ft.-° F.

The results were correlated within $\pm$ 18%, and with an average deviation of 7%, by the dimensional equation

$$h = 87\ (\sin \phi)^{0.2} (\Gamma)^{1/3}$$

for a mean-film temperature of 120°F.

RECOMMENDATIONS

Recommendations for future work on liquid side heat transfer coefficients for an inclined falling-film heater are as follows:

1. The development of a more satisfactory method for obtaining complete gravity flow. This may best be accomplished by using some type of weir. If any type of spray distributor is used it is recommended that a longer calming section which is flush with the heating surface be used.
2. The development of a system for investigating the effect of the angle of inclination between zero and twenty degrees. At some angle in this range, the heat transfer coefficient ceases to be a function of $(\sin \phi)^{0.2}$.
3. The determination of the effect of the angle of inclination on heat transfer coefficients as the liquid moves in streamline motion. This may best be accomplished by using some type of oil with a high viscosity.

NOMENCLATURE

A	Area of heat transfer surface, square feet.
c	Specific heat, Btu./lb.-° F.
C	Constant.
g	Acceleration of gravity, feet/(hour)^2.
h	Observed liquid side heat transfer coefficient, Btu./hr.-sq.ft.-° F.
h_c	Calculated liquid side heat transfer coefficient (from correlating equation), Btu./hr.-sq.ft.-° F.
k	Thermal conductivity, Btu./hr.-sq.ft.- (° F. per foot).
K	Constant.
L	Thickness, feet.
p	Density, pounds per cubic foot.
q	Heat transferred, Btu./hr.
$(Re)_m = \frac{4\Gamma}{u_m}$	Reynolds number, dimensionless.
t_i	Liquid inlet temperature, ° F.
t_o	Liquid outlet temperature, ° F.
T_a	Average thermocouple temperature, ° F.
T_p	$T_a - (\Delta T)_c$ --Temperature of plate surface, ° F.
u	Absolute viscosity, lb./hr./ft.; ordinarily, u is evaluated at bulk temperature of the stream, u_f evaluated at film temperature, u_i evaluated at t_i, u_o evaluated at t_o; $u_m = (u_i + u_o)/2$.
w	Liquid flow rate, lbs./hr.
Γ	Liquid flow rate per unit bredth, lbs./hr/ft.
$(\Delta T)_c$	Temperature correction (difference between plate surface temperature and average thermocouple temperature), ° F.
$(\Delta T)_l$	Log mean temperature difference between plate and liquid, ° F.
ϕ	Angle of inclination with respect to the horizontal, degrees.

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APPENDIX

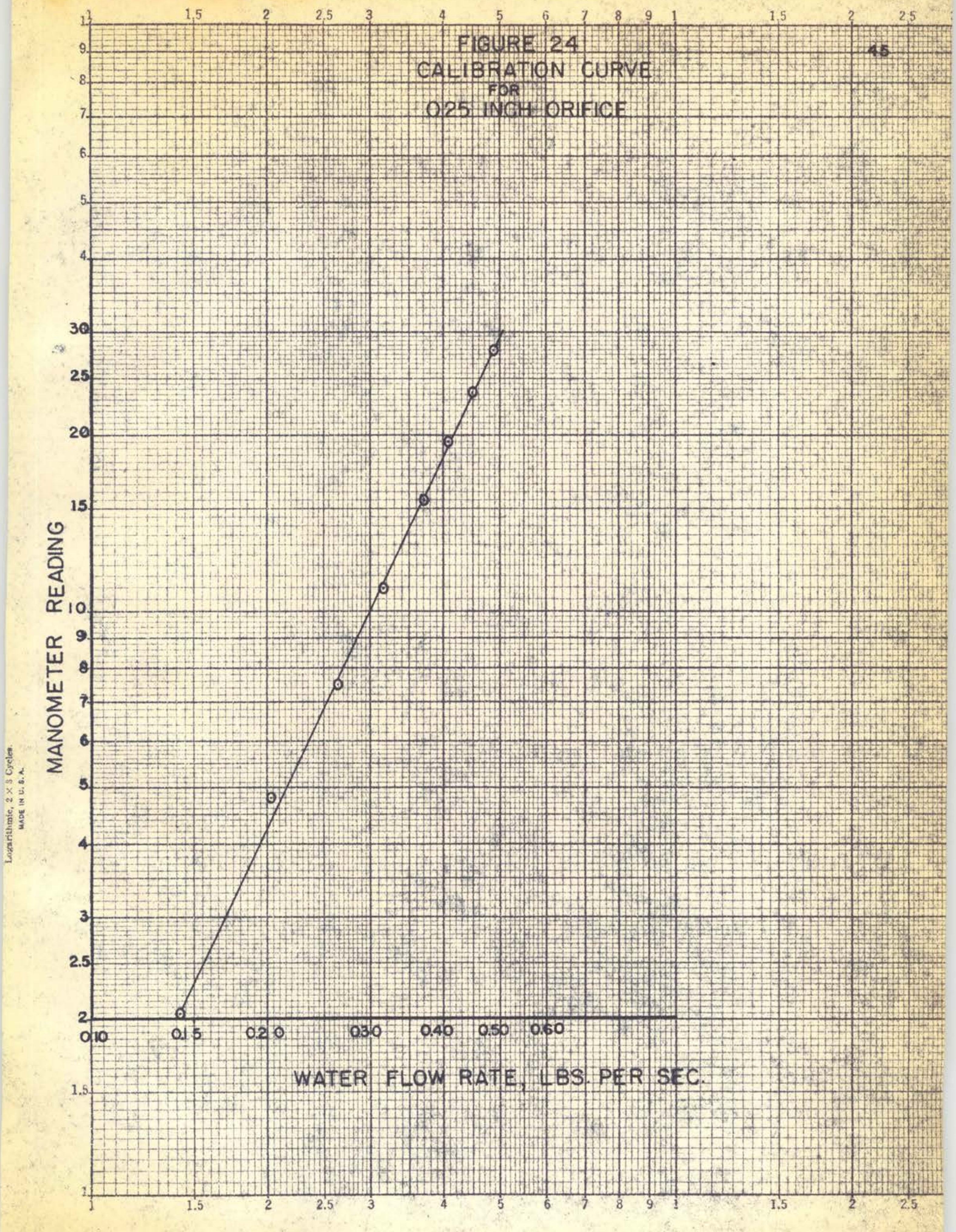

FIGURE 24
CALIBRATION CURVE
FOR
0.25 INCH ORIFICE

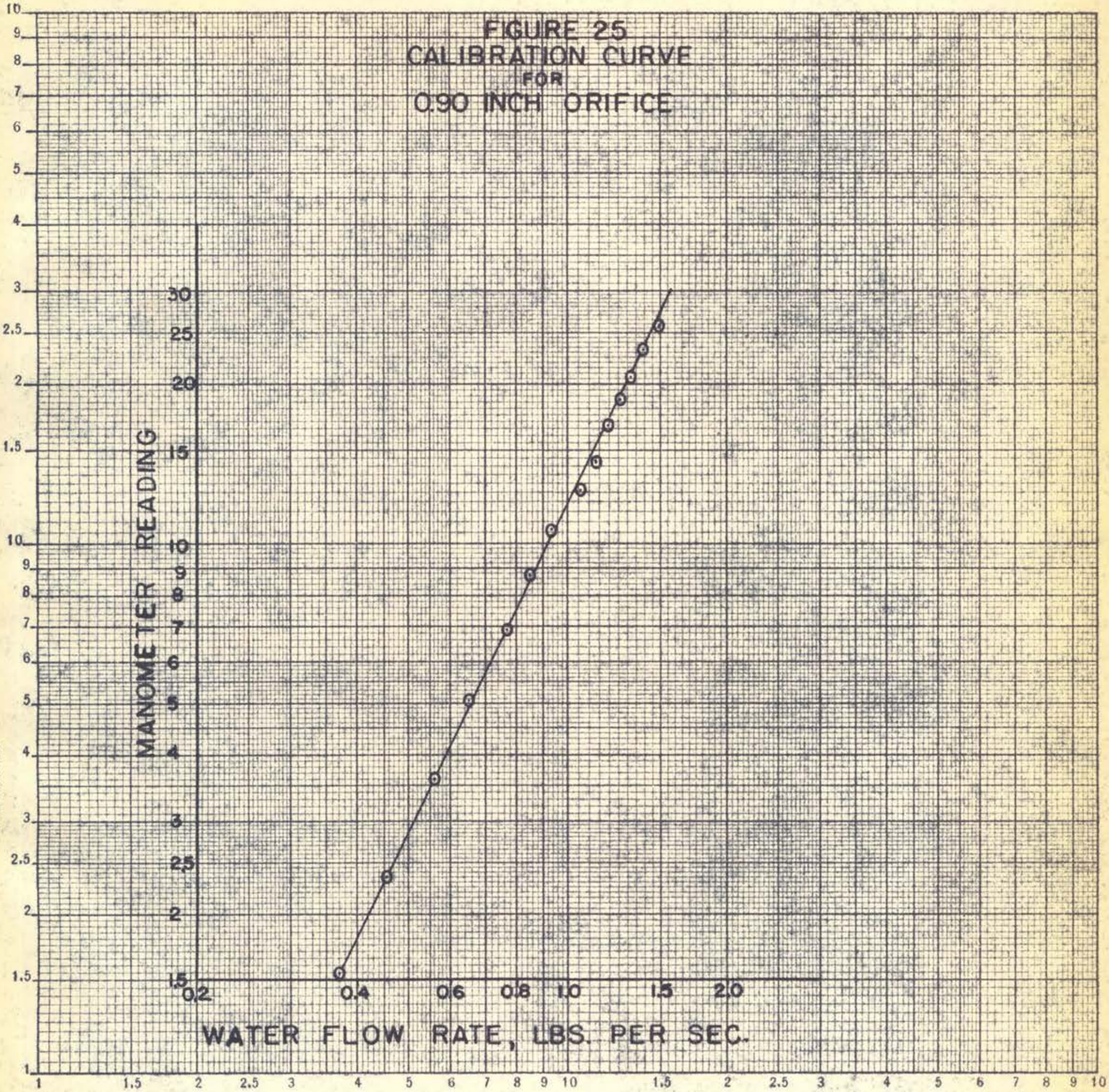
FIGURE 25
CALIBRATION CURVE
FOR
0.90 INCH ORIFICE
MANOMETER READING
30
25
20
15
10
9
8
7
6
5
4
3
2.5
2
1.5
0.2
0.4
0.6
0.8
1.0
1.5
2.0
WATER FLOW RATE, LBS. PER SEC.

TABLE I

Orifice: 1/4"
Distributor: No. 1
Area: 1.325 ft^2

Run	No.	ϕ	w	Γ	t_i in	t_o out	t_o-t_i	q	$(\Delta T)_c$	T_1	T_2	T_3	T_4	T_a	T_p	$(\Delta T)_l$ log mean	h	$(\sin\phi)^{0.2}(\Gamma)^{1/3}$	$\frac{h}{(\sin\phi)^{0.2}}$	h_c	h_c-h	% Deviation
I	A	67	1040	1850	42	110	68	70,600	2.4	135	150	151	150	146.5	144.1	62.0	867	12.1	880	1050	+183	+17.4
	B	56	1040	1850	42	112	70	72,800	2.5	134	150	160	150	148.5	146.0	62.5	875	11.8	910	1030	+155	+15.0
	C	42	1040	1850	42	114	72	74,800	2.6	133	153	163	151	150.0	147.4	62.7	896	11.3	970	980	+ 84	+ 8.6
	D	28	1040	1850	42	117	73	75,900	2.6	133	151	154	155	148.3	145.7	57.7	988	10.5	1150	910	- 78	- 8.6
II	A	18	1224	1945	44	103	59	73,200	2.5	139	150	154	149	148.0	145.5	67.5	817	9.85	1040	860	+ 43	+ 5.0
	B	28	1224	1945	43	106	63	77,200	2.7	137	148	152	138	143.8	141.1	64.4	902	10.7	1050	930	+ 28	+ 3.0
	C	42	1224	1945	43	104	61	74,600	2.6	133	142	153	137	141.2	138.6	59.8	938	11.5	1020	1000	+ 62	+ 6.2
	D	56	1224	1945	43	105	62	76,000	2.6	133	147	149	142	144.0	141.4	62.1	918	12.0	950	1040	+122	+11.7
	E	67	1224	1945	43	104	61	74,600	2.6	132	146	139	142	139.8	137.2	58.5	959	12.3	980	1070	+111	+10.4
III	A	67	1335	2425	41	101	60	80,000	2.8	125	137	137	130	132.2	129.4	52.9	1138	13.2	1160	1150	+ 12	+ 1.0
	B	56	1335	2425	41	100	59	78,700	2.7	126	138	137	132	133.2	130.5	54.9	1078	13.0	1120	1130	+ 52	+ 4.6
	C	42	1335	2425	41	102	61	81,400	2.8	129	140	137	131	134.2	131.4	54.4	1126	12.4	1220	1080	- 46	- 4.3
	D	28	1335	2425	41	101	60	80,000	2.8	131	143	142	132	137.0	134.2	58.0	1039	11.5	1210	1000	- 39	- 3.9
	E	18	1335	2425	41	100	59	78,700	2.7	131	145	150	130	139.0	136.3	61.2	967	10.6	1230	920	- 47	- 5.1
IV	A	18	1631	2900	41	89	48	78,300	2.7	126	136	136	127	131.2	128.5	60.1	978	11.3	1240	980	+ 2	+ 0.2
	B	28	1631	2900	41	90	49	79,900	2.8	124	134	131	124	128.2	125.4	56.5	1063	12.2	1240	1060	- 3	- 0.3
	C	42	1631	2900	41	91	50	81,500	2.8	124	132	132	125	128.2	125.4	55.5	1104	13.1	1200	1140	+ 36	+ 3.2
	D	56	1631	2900	41	92	51	83,100	2.9	123	132	132	125	128.0	125.1	54.7	1142	13.7	1190	1190	+ 48	+ 4.0
	E	67	1631	2900	41	91	50	81,500	2.8	122	129	129	125	126.2	123.4	53.7	1142	14.0	1160	1220	+ 78	+ 6.4
V	A	67	1811	3180	41	88	47	83,900	2.9	120	122	129	124	123.8	120.9	53.0	1190	14.5	1210	1260	+ 70	+ 5.5
	B	56	1808	3200	40	87	47	84,500	2.9	121	124	129	124	124.5	121.6	54.6	1163	14.2	1210	1240	+ 77	+ 6.2
	C	42	1800	3200	40	86	46	82,700	2.9	121	124	129	124	124.5	121.6	55.5	1121	13.6	1220	1180	+ 59	+ 5.0
	D	28	1800	3215	40	85	45	81,300	2.8	121	125	130	124	125.0	122.2	56.7	1078	12.6	1260	1100	+ 22	+ 2.0
	E	18	1786	3220	40	84	44	79,700	2.8	123	127	133	125	127.0	124.2	59.4	1010	11.6	1280	1010	0	0

TABLE II

Orifice: 0.9"
Distributor: No. 1
Area: 1.325

Run	No.	ϕ	w	Γ	t_i	t_o	$t_o - t_i$	q	$(\Delta T)_c$	T_1	T_2	T_3	T_4	T_a	T_p	$(\Delta T)_1$	h	$(\sin \phi)^{0.2}(\Gamma)^{1/3}$	$\frac{h}{(\sin \phi)^{0.2}}$	h_c	$h_c - h$	% Deviation
VI	A	18	1836	3265	41	83	42	77,100	2.7	116	120	123	118	119.2	116.5	51.6	1124	11.7	1420	1020	-104	-10.2
	B	28	1836	3265	41	85	44	80,700	2.8	113	117	120	115	116.2	113.4	47.0	1292	12.7	1510	1110	-182	-16.4
	C	42	1836	3265	41	85	44	80,700	2.8	111	114	118	114	114.2	111.4	45.0	1350	13.7	1460	1190	-160	-13.4
	D	56	1836	3265	41	85	44	80,700	2.8	112	114	120	116	115.5	112.7	46.2	1313	14.3	1370	1240	- 73	- 5.9
	E	67	1836	3265	42	86	44	80,700	2.8	112	113	117	116	114.5	111.7	44.0	1380	14.6	1400	1270	-110	- 8.7
	F	28	1836	3265	42	86	44	80,700	2.8	114	118	121	116	117.2	114.4	47.0	1292	12.7	1510	1110	-182	-16.4
VII	A	18	2160	3840	41	78	37	79,800	2.8	113	119	126	115	118.2	115.4	53.8	1117	12.4	1410	1080	- 37	- 3.4
	B	28	2160	3840	41	80	39	84,100	2.9	112	117	125	112	116.5	113.6	50.6	1250	13.4	1460	1170	- 80	- 6.8
	C	42	2160	3840	41	80	39	84,100	2.9	109	113	125	109	114.0	111.1	48.0	1320	14.4	1430	1250	- 70	- 5.6
	D	56	2160	3840	41	81	40	86,400	3.0	109	111	124	109	113.2	110.2	46.4	1402	15.1	1460	1310	- 92	- 7.0
	E	67	2160	3840	41	82	41	88,500	3.1	109	112	123	110	113.5	110.4	46.0	1450	15.4	1480	1340	- 90	- 6.7
	F	18	2160	3840	41	78.5	37.5	81,000	2.8	112	117	126	112	116.8	114.0	52.0	1172	12.4	1490	1080	- 92	- 8.5
VIII	A	18	2540	4510	41	74.5	33.5	85,000	2.9	107	110	123	111	112.8	109.9	50.2	1272	13.0	1610	1130	-142	-12.6
	B	28	2520	4480	41	76	35	88,200	3.0	105	108	120	115	112.0	109.0	48.4	1372	14.1	1600	1230	-142	-11.5
	C	42	2520	4480	41	76	35	88,200	3.0	105	108	120	114	111.8	108.8	48.0	1384	15.2	1500	1320	- 64	- 4.8
	D	56	2520	4480	41	77	36	90,600	3.1	104	106	119	122	112.8	109.7	48.4	1410	15.9	1470	1380	- 30	- 2.2
	E	67	2500	4450	41	78	37	92,400	3.2	106	106	119	116	111.8	108.6	46.5	1492	16.2	1520	1410	- 82	- 5.8
IX	A	18	2880	5120	40	69	29	83,500	2.9	104	101	111	104	105.0	102.1	45.9	1369	13.6	1730	1180	-189	-16.0
	B	28	2880	5120	40	70.5	30.5	87,700	3.0	103	100	110	103	104.0	101.0	44.2	1492	14.8	1740	1280	-212	-16.6
	C	42	2880	5120	40	71	31	89,200	3.1	102	101	110	104	104.2	101.1	43.8	1531	15.9	1660	1380	-151	-10.9
	D	56	2880	5120	40	72	32	92,100	3.2	105	104	113	107	107.2	104.0	46.4	1497	16.6	1560	1450	- 47	- 3.2
	E	67	2880	5120	40	73	33	95,000	3.3	108	104	114	109	108.8	105.5	46.9	1524	16.9	1550	1470	- 54	- 3.7
X	A	18	3180	5660	40	67	27	85,900	3.0	107	101	112	107	106.8	103.8	49.1	1312	14.1	1670	1230	- 82	- 6.9
	B	28	3180	5660	40	68	28	89,000	3.1	106	99	109	103	104.2	101.1	46.6	1437	15.3	1670	1330	-107	- 8.0
	C	42	3180	5660	40	69	29	92,200	3.2	104	97	107	100	102.0	98.8	42.6	1626	16.4	1760	1430	-196	-13.7
	D	56	3180	5660	40	70	30	95,400	3.3	109	100	109	104	105.5	102.2	45.4	1580	17.1	1640	1490	- 90	- 6.0
	E	67	3180	5660	40	70	30	95,400	3.3	112	99	107	107	106.2	102.9	46.3	1551	17.5	1580	1520	- 31	- 2.0
XI	A	18	3675	6520	40	66	26	95,500	3.3	109	97	108	112	106.5	103.2	49.0	1470	14.7	1860	1280	-190	-14.8
	B	28	3675	6520	40	66.5	26.5	97,400	3.4	107	94	104	110	103.8	100.4	46.1	1588	16.0	1850	1390	-198	-14.3
	C	42	3675	6520	40	67	27	99,100	3.4	107	93	106	111	104.2	100.8	46.9	1588	17.2	1720	1500	- 88	- 5.9
	D	56	3675	6520	40	68	28	103,000	3.6	110	95	104	113	105.5	101.9	46.5	1667	17.9	1730	1560	-107	- 6.9
	E	67	3640	6460	40	69	29	105,600	3.6	113	92	100	111	104.0	100.4	44.3	1790	18.3	1820	1590	-200	-12.6

TABLE III

Orifice: 0.9"
Distributor: No. 2

Run	No.	ø	w	Γ	t_i	t_o	t_o-t_i	q	A	$(\Delta T)_c$	T_1	T_2	T_3	T_4	T_a	T_p °F	$(\Delta T)_1$	h	$(\sin ø)^{0.2}(\Gamma)^{1/3}$	$\frac{h}{(\sin ø)^{0.2}}$	h_c	h_c-h	% Deviation
XII	A	9	4015	7120	40	60	20	80,300	1.313	2.8	116	122	111	110	114.8	112.0	61.4	997	13.3	1450	1160	+163	+14.1
	B	18	4015	7120	41	62.5	21.5	86,500	1.307	3.0	116	119	109	102	111.5	108.5	55.7	1212	15.2	1540	1320	+108	+ 8.2
	C	28	4015	7120	41	64	23	92,500	1.301	3.2	116	116	107	101	110.0	106.8	53.3	1337	16.5	1560	1440	+103	+ 7.1
	D	42	4015	7120	42	65.5	23.5	94,500	1.295	3.3	111	110	104	98	105.8	102.5	47.6	1536	17.6	1660	1530	- 6	- 0.4
	E	56	4015	7120	42	66.5	24.5	98,500	1.289	3.4	109	109	103	98	104.8	101.4	46.0	1663	18.5	1730	1610	- 53	- 3.3
	F	67	4015	7120	41	66.5	25.5	102,400	1.283	3.5	158*	108	101	99	102.7	99.2	44.1	1810	18.9	1840	1640	-170	-10.4
	G	90	4015	7120	41	65	24	96,500	1.254	3.3	221*	105	105	103	104.3	101.0	47.0	1639	19.2	1640	1670	+ 31	+ 1.9
XIII	A	9	4260	7580	43	62.5	19.5	83,200	1.307	2.9	116	119	122	120	119.2	116.3	62.7	1017	13.5	1470	1170	+153	+13.1
	B	18	4260	7580	43	64	21	89,500	1.301	3.1	113	115	110	120**	112.7	109.6	55.3	1245	15.5	1580	1350	+105	+ 7.8
	C	28	4260	7580	43	64.5	21.5	92,700	1.295	3.2	113	110	110	120**	111.0	107.8	53.1	1350	16.8	1570	1460	+110	+ 7.5
	D	42	4260	7580	43	65	22	93,800	1.289	3.2	111	108	108	116**	109.0	105.8	50.8	1436	18.1	1560	1570	+134	+ 8.5
	E	56	4260	7580	43	65.5	22.5	96,000	1.283	3.3	129*	105	104	115**	104.5	101.2	46.0	1627	18.9	1690	1640	+ 13	+ 0.8
	F	67	4260	7580	43	65.5	22.5	96,000	1.278	3.3	174*	104	103	114**	103.5	100.2	44.9	1677	19.3	1710	1680	+ 3	+ 0.2
	G	90	4260	7580	43	65.0	22	93,800	1.254	3.2	213*	104	103	115**	103.5	100.3	45.4	1650	19.6	1650	1700	+ 50	+ 2.9
XIV	A	9	4760	8470	43	60	17	81,000	1.301	2.8	115	115	119	129**	116.3	113.5	61.2	1018	14.0	1480	1220	+202	+16.5
	B	18	4760	8470	43	61	18	85,700	1.295	3.0	111	111	111	122**	111.0	108.0	55.6	1190	16.1	1510	1400	+210	+15.0
	C	28	4760	8470	43	62	19	90,500	1.289	3.1	107	105	106	115**	106.0	102.9	49.5	1414	17.5	1650	1520	+106	+ 7.0
	D	42	4760	8470	43	63	20	95,200	1.283	3.3	116*	104	104	114**	104.0	100.7	46.8	1588	18.8	1720	1640	+ 52	+ 3.2
	E	56	4760	8470	42.5	62.5	20	95,200	1.278	3.3	158*	103	103	119**	103.0	99.7	46.3	1611	19.6	1670	1700	+ 89	+ 5.2
	F	67	4750	8450	42	63.5	21.5	102,200	1.272	3.5	182*	102	102	115**	102.0	98.5	44.7	1796	20.0	1830	1740	- 56	- 3.2
	G	90	4760	8470	42	62.5	20.5	97,500	1.243	3.4	202*	103	103	110**	103.0	99.6	46.4	1692	20.4	1690	1770	+ 78	+ 4.4
XV	A	9	5165	9180	42	57.5	15.5	80,100	1.295	2.8	109	109	111	101	107.5	104.7	54.4	1140	14.4	1650	1250	+110	+ 8.8
	B	18	5050	9000	42	58.5	16.5	83,300	1.289	2.9	107	107	104	97	103.8	100.9	50.1	1290	16.4	1630	1430	+140	+ 9.8
	C	28	5050	9000	42	59.5	17.5	88,400	1.283	3.1	109	104	103	95	102.8	99.7	48.9	1410	17.8	1640	1550	+140	+ 9.0
	D	42	5050	9000	42	60.5	18.5	93,500	1.278	3.2	143*	102	99	92	97.7	94.5	42.5	1723	19.0	1870	1650	- 73	- 4.4
	E	56	5050	9000	42	60.5	18.5	93,500	1.272	3.2	170*	102	99	94	98.3	95.1	42.6	1728	20.0	1790	1740	+ 12	+ 0.7
	F	67	5050	9000	42	61.0	19.0	96,000	1.266	3.3	182*	103	99	96	99.3	96.0	43.8	1734	20.4	1760	1770	+ 36	+ 2.0
	G	90	4960	8840	42	60.0	18.0	89,300	1.231	3.1	202*	105	102	95	100.7	97.6	45.8	1583	20.8	1580	1810	-227	-12.5
XVI	A	9	5785	10120	42	57	15	86,800	1.289	3.0	105	105	108	102	105.0	102.0	52.1	1292	14.9	1870	1300	+ 8	+ 0.6
	B	18	5785	10120	42	58	16	92,500	1.283	3.2	108	104	104	99	103.8	100.6	50.0	1442	17.1	1830	1490	+ 48	+ 3.2
	C	28	5785	10120	42	59	17	98,300	1.278	3.4	128*	102	101	96	99.7	96.3	45.1	1705	18.5	1990	1610	- 95	- 5.9
	D	42	5785	10120	42	60	18	104,000	1.272	3.6	160*	104	100	100	101.3	97.7	45.8	1785	19.9	1930	1730	- 55	- 3.2
	E	56	5765	10100	42	60	18	103,800	1.266	3.6	170*	100	96	99	98.3	94.7	42.9	1912	20.8	1990	1810	-102	- 5.6
	F	67	5785	10120	42	60	18	104,000	1.260	3.6	184*	102	96	103	100.3	96.7	44.9	1840	21.2	1870	1840	0	0
	G	90	5615	9980	42	60	18	101,000	1.219	3.5	202*	103	102	101	102	98.5	46.7	1775	21.6	1880	1880	+105	+ 5.6

* Thermocouple reading not used in average since liquid hitting further down the plate than point where thermocouple was located.

** Thermocouple reading not used in average.

SAMPLE CALCULATIONS

Flow rate:

0.453 lb./sec. as read for 24.2 inches of carbon tetrachloride minus water (see Figure 21).

$0.453 \times 3600 = 1631$ lbs./hr.

$$\Gamma = \frac{1631}{6.75/12} = 2900 \text{ lbs./hr./ft.}$$

Heat transferred:

$$q = (w)(c)(t_o - t_i)$$
$$= 1631(1)(89 - 41)$$
$$= 78{,}300 \text{ Btu./hr.}$$

Estimated distance thermocouples are below surface of plate is 0.05 inch for three of the thermocouples, one couple being on the surface. Average estimated distance below surface is:

$$L = 3(0.05)/4 = 0.0375 \text{ inch or } 0.00313 \text{ ft.}$$

Temperature correction:

$$q = (k/L)(A)(\Delta T)_c$$
$$78{,}300 = (68/0.00313)(1.325)(\Delta T)_c$$
$$(\Delta T)_c = 2.7\ ^{\circ}\text{F.}$$

Average thermocouple temperature:

$$T_a = \frac{T_1 + T_2 + T_3 + T_4}{4} = \frac{126 + 136 + 136 + 127}{4} = 131.2^{\circ}\text{ F.}$$

Temperature of plate:

$$T_p = T_a - (\Delta T)_c$$
$$= 131.2 - 2.7 = 128.5^{\circ}\text{ F.}$$

Log mean temperature difference between plate and fluid:

$$(\Delta T)_1 = \frac{(T_p - t_i) - (T_p - t_o)}{\ln \dfrac{T_p - t_i}{T_p - t_o}}$$

$$(\Delta T)_1 = \frac{t_o - t_i}{\ln \dfrac{T_p - t_i}{T_p - t_o}}$$

$$= \frac{89 - 41}{\ln \dfrac{128.5 - 41}{128.5 - 89}}$$

$$= \frac{48}{\ln 2.22}$$

$$= 48/0.799$$

$$= 60.1^{\circ}\ F.$$

Heat transfer coefficient (from data):

$$q = h(A)(\Delta T)_1$$

$$78{,}300 = (h)(1.325)(60.1)$$

$$h = 978\ \text{Btu./hr.-sq.ft.-}^{\circ}\ F.$$

Heat transfer coefficient (from correlating equation):

$$h_c = 87(\sin \phi)^{0.2}\ (\Gamma)^{1/3}$$

$$= 87(\sin 18^{\circ})^{0.2}\ (\Gamma)^{1/3}$$

$$= 87(0.309)^{0.2}(2900)^{1/3}$$

$$= 980\ \text{Btu./hr.-sq.ft.-}^{\circ}\ F.$$

Deviation:

$$h_c - h = 980 - 978 = +\ 2\ \text{Btu./hr.-sq.ft.-}^{\circ}\ F.$$

% deviation:

$$(h_c - h)/h_c\ (100) = (2/980)(100) = +\ 0.2\%$$

Reynolds Number:

$$u_m = \frac{u_i + u_o}{2} = \frac{3.73 + 1.45}{2} = \frac{5.18}{2} = 2.59\ \text{lbs./hr./ft.}$$

$$Re = \frac{4\Gamma}{u_m} = \frac{4(1850)}{2.59} = 2900$$

$$u_m = \frac{u_i + u_o}{2} = \frac{3.71 + 2.71}{2} = \frac{6.42}{2} = 3.21\ \text{lbs./hr/ft.}$$

$$Re = \frac{4\Gamma}{u_m} = \frac{4(10{,}120)}{3.21} = 12{,}800$$

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